TECHNICAL UNIVERSITY OF CRETE
SCHOOL OF MINERAL RESOURCES ENGINEERING
MSc in Petroleum Engineering

MSc Thesis

Design of oil, gas and mud pumps during well drilling and reservoir production

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Abstract

Pumps are of high importance in all sectors of oil and gas industry. Starting from the first stages in the development of a field, the lack of pumps would make the drilling process impossible as they are used to circulate drilling mud so as to keep the formation fluids under control, bring the cutting to the surface and cool the drilling bit. During production pumps are useful when artificial lift is needed. There is a variety of pumps to choose from depending on the specific characteristics of each reservoir fluid and oil filed. There are pumps that fit better for onshore or for offshore use varying from big heavy installations to light delicate super-tech devices. Pumps may be used to produce at the surface various fluid rates. The variety of the produced reservoir fluid properties and conditions lead to a variety of pumps capable of lifting viscous fluid, rich in solid or gas content, higher temperature fluids etc. Combinations of pumps developed and new hybrid pumps, designed to meet the upcoming needs of oil and gas industry, are also available in the market. Pumps are also present in fracturing operations where they provide the huge hydraulic power necessary at surface to fracture the rock segment. Last but not least important use is the products transfer of the separated and treated fluids from the surface to the final sales point.

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Chapter 1 Types of Pumps

1.1 Positive Displacement Pump (PDP)

A positive displacement pump is a device that moves fluids and slurries by rotary or reciprocating operating mechanisms.
As pump can be described the part from the suction flange to the discharge flange and is being differentiated from the “system” which includes the motor, coupling, baseplate, tanks, connecting piping and controls. A positive displacement pump can be thought of as a "flow generator", as compared to a centrifugal pump, which can be imagined as a "pressure or head generator".

1.1.1 Definitions of Major Terms and Nomenclature

Fluid versus Liquid

A fluid is a substance that is unable to resist even the slightest amount of shear without flowing. Liquids, gases and vapors are fluids, but liquids are practically incompressible, with an exception at very high pressures.

Flow or "Capacity"

Flow is the volume of fluid per unit of time. The ideal flow disregards "slip" or the portion of flow that "slips" back through the internal clearances from discharge side to suction, driven by the differential pressure across the pump. The net flow is what actually comes out from the pump exit into the discharge pipe.

Pressure

There are discharge, suction and differential pressure terms. It is very important to specify gauge (psig), absolute (psia), or differential (psid) pressure. Actual available suction pressure must be greater than the minimum required suction pressure, in order to avoid adverse effects on pump operation, such as flow loss, cavitation, noise and vibration.

Net Positive Inlet Pressure (NPIP)

NPIP is the pump inlet pressure minus the fluid vapor pressures. Positive displacement pumps typically operate in systems with low inlet velocities and in these cases, the velocity head portion (dynamic head) has historically been ignored. Inlet conditions for positive displacement pumps have been defined in pressure terms.

NPIPR with the "R" signifying "pump required" and is the difference between the inlet pressure and the vapor pressure—corrected to the centerline of the pump inlet port— which is necessary for the pump to operate without a reduction in flow. The Hydraulic Institute defines the minimum required pressure (or equivalent NPIPR) as such where 5% of the flow reduction occurs due to cavitation.

NPIPA with the "A" signifying "pump available" for positive displacement pumps. Any increase in NPIPA has no effect on volumetric efficiency, as long as it is above the NPIPR. Low values of NPIPA may result not only in flow reduction but also in significant pressure spikes, vibrations, noise and possible damage to the pump.

Power

Power is defined as energy per unit of time. The gross power delivered by the driver to the pump is called *break horsepower* (BHP).The net power delivered to the fluid by the pump is called *hydraulic* or *fluid power* (FHP) and can be calculated as the product of the differential pressure times the theoretical flow with the appropriate coefficient, plus internal viscous power losses. The difference between the break and hydraulic power is due to the internal losses or the sum of mechanical and volumetric losses.

Efficiency

Volumetric efficiency has been a much more common method of comparison between different designs and applications. Positive displacement pumps are inherently constant flow machines with regard to differential pressure. In theory, a constant volume of fluid is "displaced" with every rotation, stroke or cycle. However, because of the internal clearances, a certain amount of fluid "slips" back from the discharge side to suction. This slip depends on the lateral and radial clearances and on the overall differential pressure which drives it. The higher the viscosity of the fluid is the more it resists to the slip. The net actual flow is the difference between the ideal or theoretical flow and the slip:

$$Q_a = Q_o - Q_{slip}$$

which can be also expressed in terms of volumetric efficiency:

$$\eta_{vol} = \frac{Q_o - Q_{slip}}{Q_o}$$

The overall efficiency is often called "mechanical efficiency" in PD pumps and is the ratio of useful hydraulic power (FHP) transmitted to the fluid exiting the pump, to a total power (BHP) absorbed by the pump:

$$\eta = \frac{FHP}{BHP}$$

Torque

Torque is a measure of the machine's ability to move the resisting loads. Power is divided by the rotating speed in rpm and the appropriate coefficient is depended on the units. For the same power, the torque is greater at lower speed.

Viscosity

Viscosity is defined as a fluid property that characterizes its ability to resist motion. This is a major parameter influencing pump operation. We can express it in two ways:

As dynamic viscosity which is a coefficient between the fluid shear rate and shear stress, measured in centipoise (cP).
And kinematic viscosity which actually is equal to dynamic viscosity divided by the specific gravity and measured in centistokes (cSt).
Pump power goes up with viscosity to overcome the internal hydraulic viscous drag. Suction conditions become more demanding with increased viscosity, reflecting the ability of the fluid to get to the pump suction port and fill its displacement mechanism (gears, screws, etc.).

Revolutions per Minute (rpms)

In the 1980's it became common and fashionable to use smaller, faster running pumps, as opposed to larger, slower running pumps for obvious cost reasons. However, with speed came trouble, as many maintenance and reliability plant personnel discovered. A faster running pump requires more suction pressure and also wears out disproportionately faster.
Today it is important to consider not only initial cost but also wear, suction requirements, vibration and so forth, to strike a balance between speed and reliability for a given flow requirements.

Extended definitions:

Q	Pump net flow, gpm
Q_O	Pump theoretical flow, disregarding slip, gpm
Q_{slip}	Pump slip, gpm.
BHP	Pump input power, hp
rpm	Pump shaft rotational speed, revolutions per minute
p_s	Pump suction pressure (usually in absolute units, psia or psig)
p_d	Pump discharge pressure (usually gauge units, psig)
Δ_p	Differential pressure (psi or psid)
v	Kinematic viscosity of the pumped fluid, centistokes (cSt)
μ	Dynamic viscosity, centipoises (cP)
p_{smin}	Minimum required suction pressure (expressed in absolute units, psia)
η_{vol}	Volumetric efficiency
η_{mech}	Mechanical efficiency
η	Overall efficiency
PF	Motor power factor
η_{motor}	Motor efficiency
T	Torque, in lbs

Flow

Convert from	Divide by	Convert to
GPM (US)	4.403	m^3/HR
GPM (US)	15.9	liters/sec

Pressure and head

Convert from	Divide by	Convert to
psi	14.7	Atm
psi	14.5	Bars
psi	145	MPa

Power

Convert from	Multiply by	Convert to
HP	0.746	KW

Figure1.1.1: Conversion table

1.1.2 Types of PDP pumps

The two major classes of positive displacement pumps are rotary and reciprocating. In the oil and gas industry the majority of applications are handled by the following pump types:

I. Rotary
   - Gear
   - Lobe
   - Screw
   - Progressive Cavity (special screw type)
   - Vane

II. Reciprocating
   - Diaphragm
   - Piston/Plunger

1.1.3 Rotary

1.1.3.1 Rotary (Gear)

Gear pump is the simplest of the rotary displacement pumps. There are two different designs based on the rotation transfer and the size of the gear.

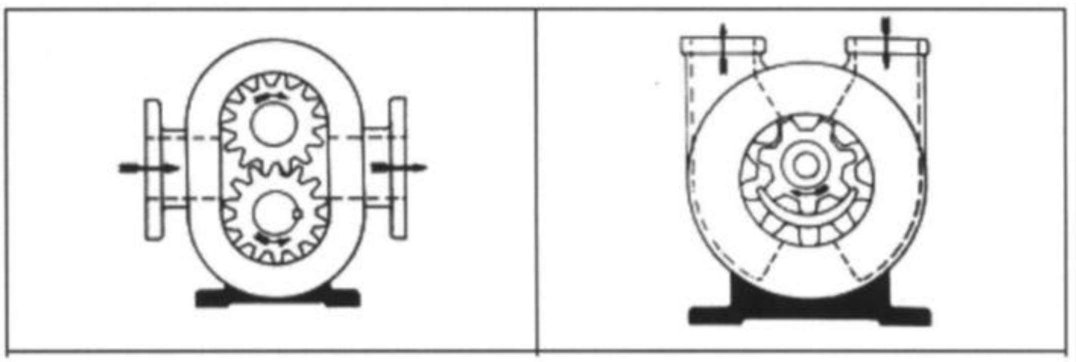


Figure1.1.3.1a: External gear pump (left) and internal gear pump (right)

Internal gear

Internal gear design consisted of two different in size gears and follows the “Gear in gear principle”. The internal gear, the smaller one, gets its drive from the engine (drive gear) and the external gear, bigger one, gets the drive from the internal gear (driven gear) as they being engaged on the upper side of the pump.
A crescent which does not rotate separates those gears and helps the fluid to get trapped between them. The positive displacement of the fluid occurs by the filling of the area between the rotor (small gear) and the idler (big gear).
The two gears rotate in the same direction and the rotation can be clockwise or counterclockwise.

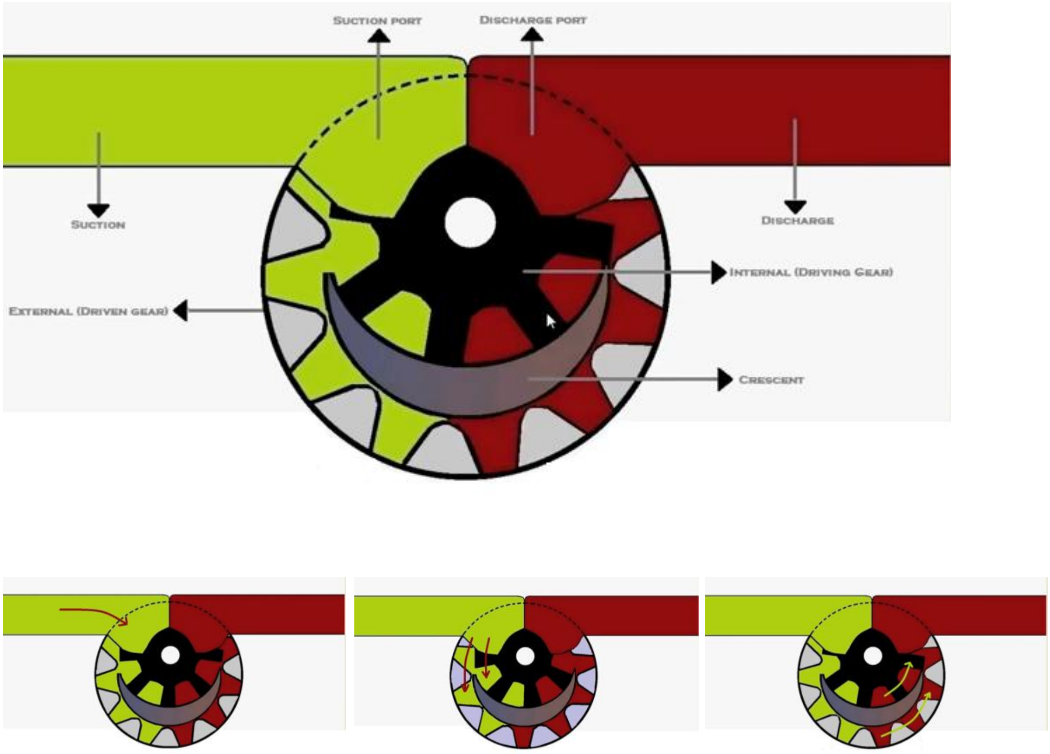


Figure1.1.3.1.b: Internal gear pump

External

External design has two similar in size gears and a whole different set-up than the internal design. They are placed one above the other so they engaged in the middle of the pump and in this way the rotation motion is being transported from the drive gear (upper) to the driven gear (lower). The gears rotate in the opposite direction with the drive gear to rotate clockwise and the driven gear to rotate counterclockwise.

The fluid that is coming from the suction side is trapped between the casing and the gears and being delivered peripherally to the gears, to the discharged side of the pump.

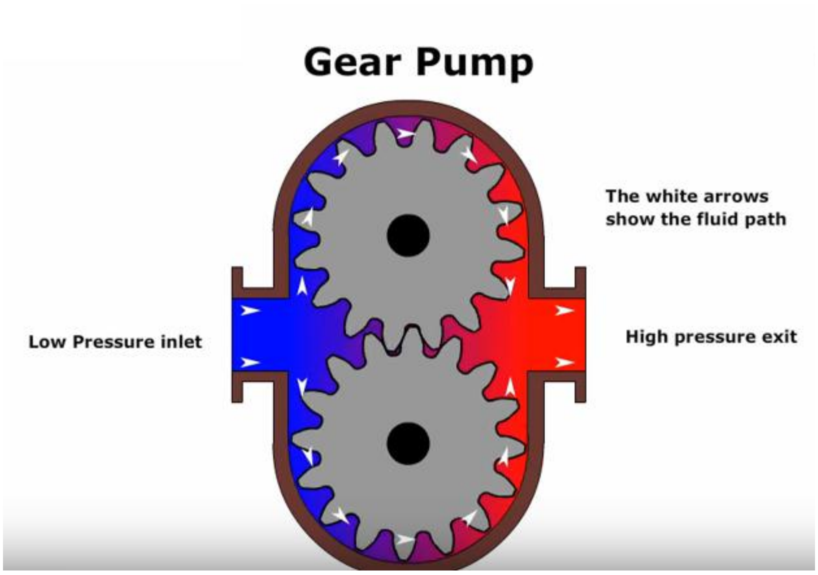


Figure1.1.3.1.c: External gear pump

1.1.3.2 Rotary (Lobe)

Lobe pumps are very similar to external gear pumps in their operation. A timing mechanism located in the gearbox is used to transfer the rotation of the drive rotor to the idler and in this way the lobes are not in contact as oppose to the external gear design.
As fluid is coming from the inlet side and the lobes come out of mesh, they create expanding volume. The liquid flows into the cavity and is trapped by the lobes as they rotate. Then it travels around the interior of the casing in the pockets between the lobes and the casing and finally, the meshing of the lobes forces liquid through the outlet port under pressure.
Different designs of lobe pumps exist and depend on the number of lobes which can vary between 1 and 5.

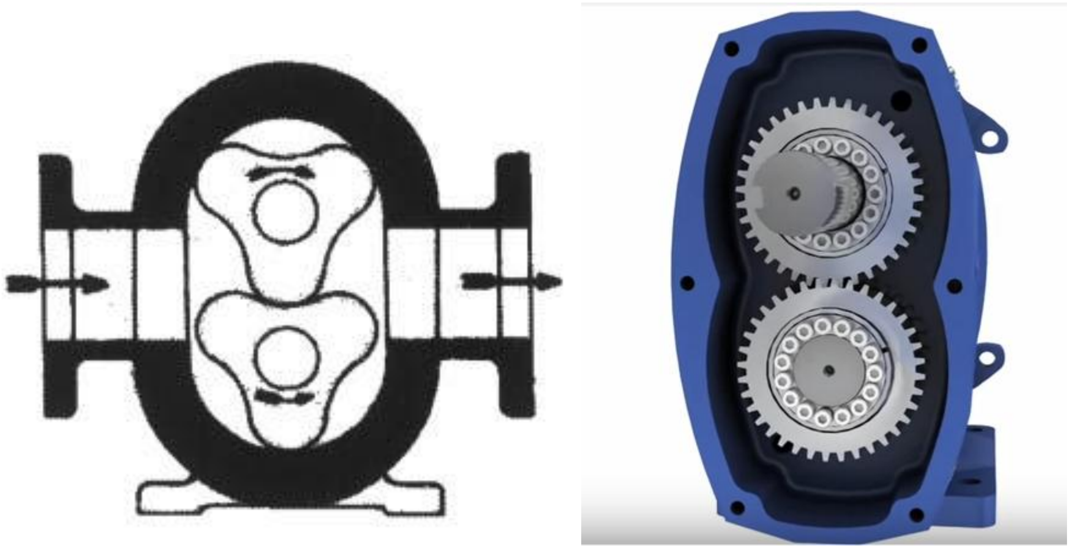


Figure 1.1.3.2.a: A three-lobe design (left) and a timing mechanism (right)

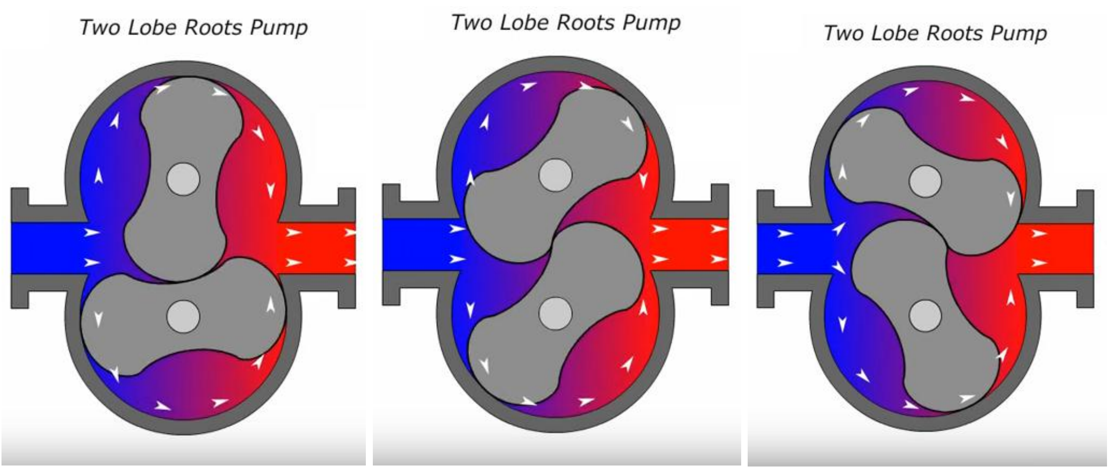


Figure1.1.3.2.b: A two-lobe design operation

1.1.3.3 Rotary (Screw)

These pumps are usually of the two-screw timed or three-screw un-timed design. In the two screw design there is a pair of spiral rotors which rotates allowing the spiral teeth to mesh together and form chambers. Inlet fluid fills these cavities between the rotors and the casing or liner.
The fluid is then moved axially to the outlet port, as the rotation moves the chambers from the intake side to the discharge side where they shrink and the fluid is being compressed.

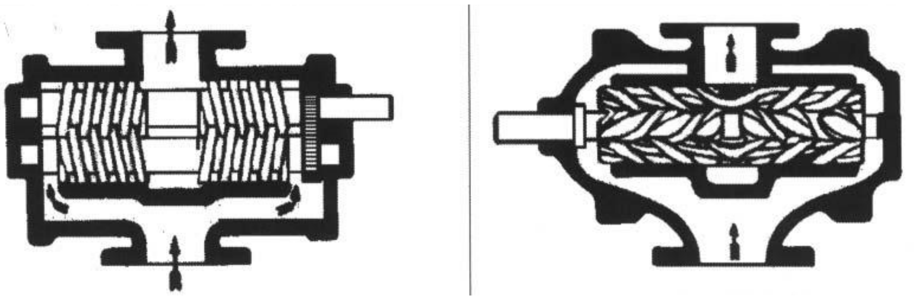


Figure 1.1.3.3.a: Two-screw pump (left) and Three-screw pump (right)

1.1.3.4 Progressive Cavity

Progressive cavity pump is a special type of screw rotary pump. The simplest progressing cavity pump consists of a helical rotor which turns inside a double-threaded nut called a stator and is ten times longer as its width.
As the rotor rotates inside the stator, two cavities form at the suction end, with one cavity opening at the same rate as the other cavity is closing. The cavities are being formed from one end of the stator to the other.
In most progressing cavity pumps the stator is made with an elastomeric material (rubber, Teflon, etc.) that fits on the rotor with a compressive fit.
This compressive fit between the rotor and stator creates seal lines where the two components contact and they can keep the cavities separated as they progress through the pump.

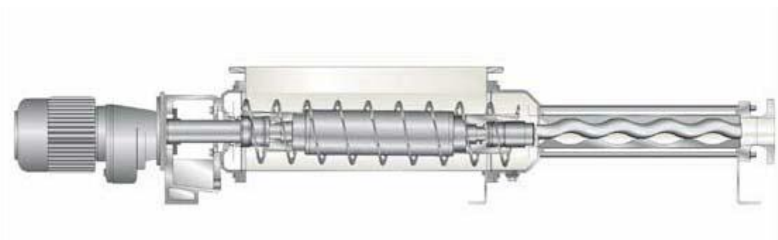


Figure1.1.3.4.a: Progressing Cavity Pump

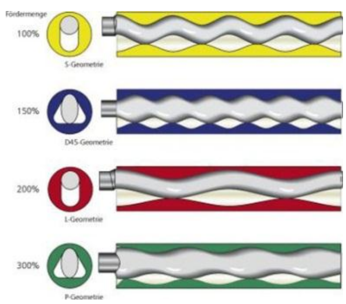


Figure1.1.3.4.b: Variety of Progressing Cavity Pumps

1.1.3.5 Rotary (Vane)

A rotary vane pump is a positive displacement pump that consists of vanes which may be in the form of blades, buckets, rollers or slippers mounted to a circular rotor that rotates inside of a large circular cavity.
The centers of these two circles are offset and cause eccentricity. In this way the vanes are allowed to slide into and out of the rotor and seal on all edges creating at the same time vane chambers.
On the intake side of the pump, the vane chambers are increasing in volume and these increasing volume vane chambers between the vane and the casing liner are filled with fluid forced in by the inlet pressure.
The fluid forms a crescent as it moves circumferentially to the outlet port where the vane chambers are decreasing in volume. Finally the fluid is forced out of the pump.

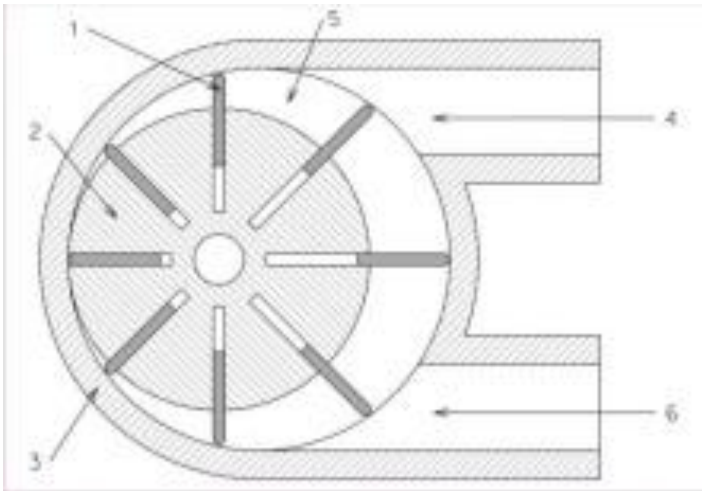


Figure1.1.3.5.a: Eccentricity of the vane pump

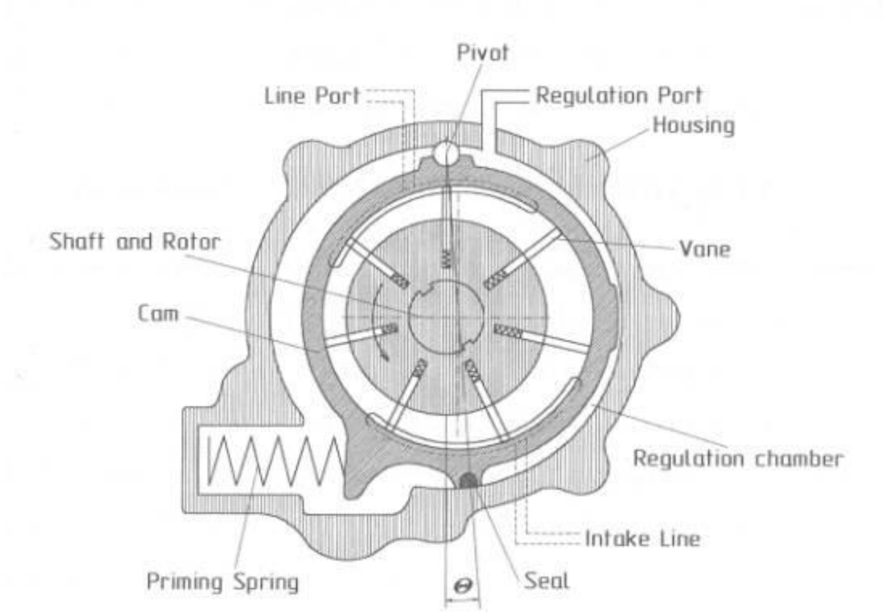


Figure1.1.3.5.b: Vane pump

1.1.4 Reciprocating

1.1.4.1 Diaphragm

Hydraulically or mechanically actuated, diaphragms displace the fluid from suction to discharge side and are aided by the valve mechanism which allows fluid to enter the pump chamber during the suction stroke and displaces it during the discharge stroke.

The liquid chambers are alternatively filled and empty by fluid drawn through a common inlet and discharged through a single outlet.

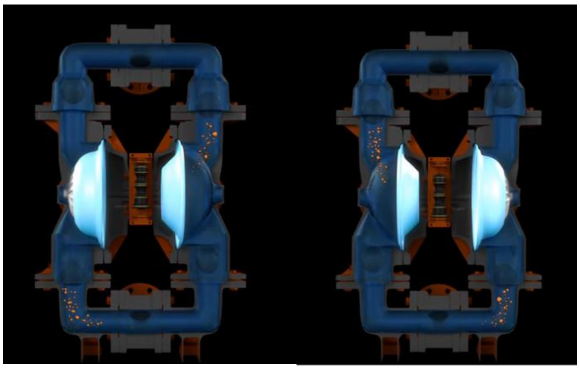


Figure 1.1.4.1.a: Liquid chambers are alternatively filled and empty by fluid

The diaphragms in each chamber are linked by a single shaft allowing them to move. The air-valve directs pressurized air to the back of diaphragm “A” and this begins chambers “B” suction stroke which starts as diaphragm “B” starts to move towards the center of the pump thereby creating a vacuum in chamber “B”.

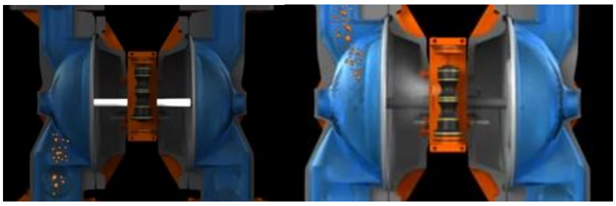


Figure1.1.4.1.b: Single shaft (left) and air-valve (right)

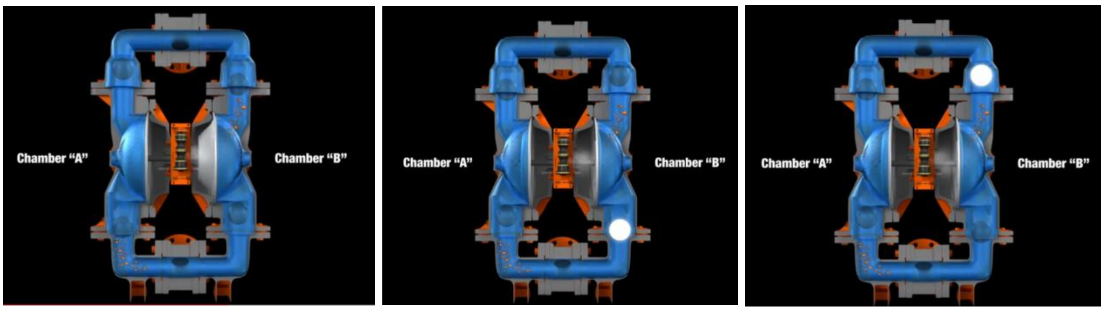


Figure1.1.4.1.c: Hydraulically actuated double diaphragm pump

Atmospheric pressure forces fluid into chamber “B” through the inlet ball valve and when the pressurized diaphragm “A” reaches the limit of its discharge stroke, the air-valve redirects pressurized air to the back of diaphragm “B”. This begins the discharge stroke of chamber “B”. The hydraulic forces developed inside of chamber “B” force the inlet ball on its seat and the discharged ball off its seat. This condition allows fluid to flow through the pumps discharge outlet. The same process occurs in the opposite chamber constituting a full cycle.

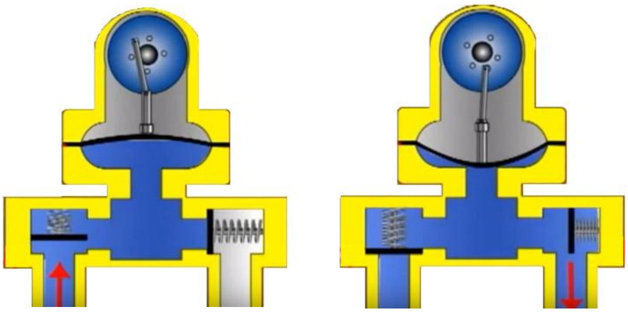


Figure 1.1.4.1.d: Mechanical actuated diaphragm pump

1.1.4.2 Piston/Plunger

A high-pressure seal reciprocates with the piston in a piston pump but for a plunger pump the high-pressure seal is stationary and a smooth cylindrical plunger slides through the seal.

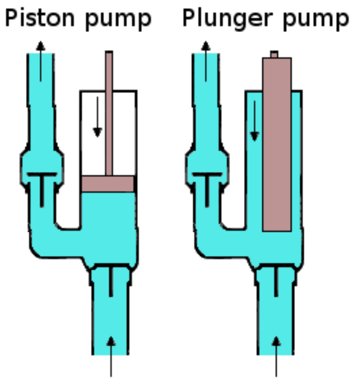


Figure1.1.4.2.a:Piston/Plunger similarities and contradictions

Piston/plunger pumps are of the reciprocating type and their working principle is the same. The liquid chamber is alternatively connected to the suction and discharge sides with a system of internal valving. The piston reciprocates inside of a cylindrical chamber aided by a rod connected to a rotor.

The discharge stroke occurs when the piston moves towards in the chamber. Pressure is created to both inlet and outlet valves which are designed to act the opposite way. The inlet valve closes and the outlet valve opens, which means that the fluid can escape and the flow starts.

In the backward motion, when the piston pulls out the chamber, the suction stroke occurs. A low pressure vacuum is created and as a result, the suction forces close the discharge valve and open the suction valve allowing fluid to pass in the pump.

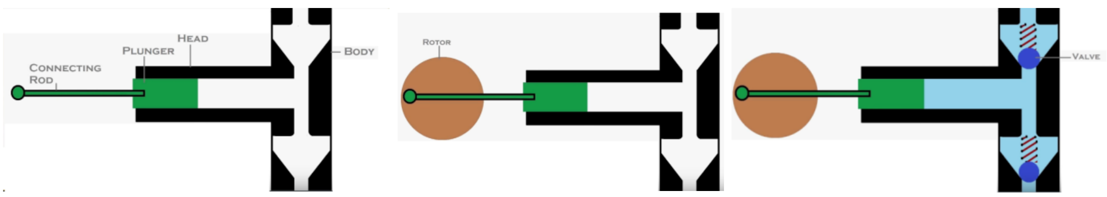


Figure1.1.4.2.b: Parts of a piston/plunger pump

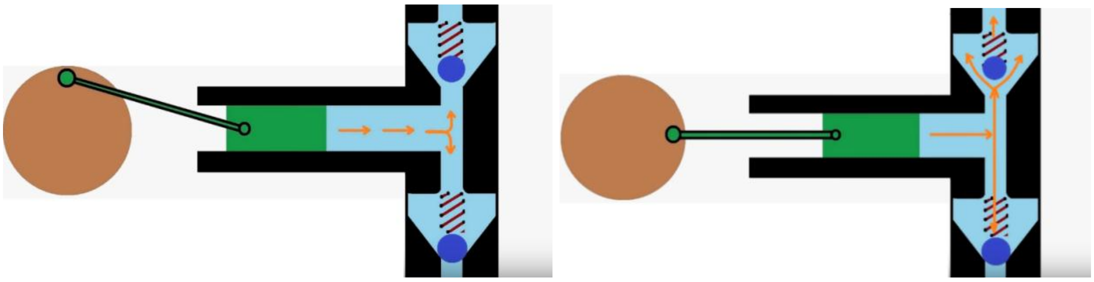


Figure1.1.4.2.c: Discharge stroke

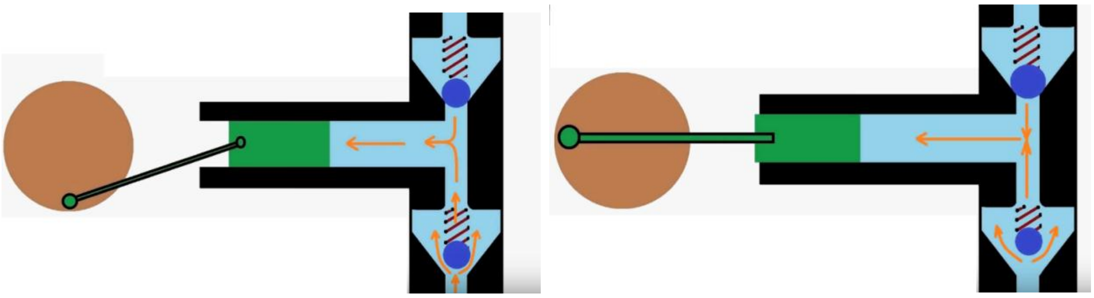


Figure1.1.4.2.d: Suction stroke

1.1.5 Tests / Measured parameters/ Preparation

Test Objective

The objective of the test is to measure pump performance at the operating point or over its range of operating points. The principal values that need to be measured are flow through the pump (Q), pump shaft speed (rpm), pressure at the discharge port (p_d) and pressure at the inlet port (p_s).

Test Equipment

Any instruments to be used in testing should have been recently calibrated. Instrument accuracy is often quoted as a percentage of full-scale reading. A 2000-psi gauge with a ±2% accuracy will measure to ± 40 psi. That is ± 4% accuracy for a measurement of 1000 psi. The same accuracy on a 5000-psi gauge is ± 100 psi to give ± 10% accuracy at 1000 psi. Gauges used should be properly sized for tested range.

Flow

For positive displacement pumps, the key measured parameter is flow. For chemical plants the most common way to measure flow is with a flow meter. Other simple methods such as, known volume vessel plus a stopwatch or other open-end measurements are not practical for the hazardous environment of a chemical plant.

For better accuracy flow meters should be periodically cleaned and calibrated. In theory, the viscosity of the measured fluid must be similar to the calibration fluid but in practice there can be a wide variation of viscosity (+/- 1000 cP) and the flow measurement error would be within + / - 2%, which is acceptable for most field tests.

Because the theoretical flow of positive displacement pumps is directly proportional to speed, another simple method is to measure the pump shaft rpm and multiply it by the pump unit flow (gallons-per-revolution). The unit flow is constant for a given pump and can be calculated from the pump performance curve at zero differential pressure. The net (actual) flow is the difference between the theoretical (zero pressure differential) flow and slip.

Slip can be obtained from the pump curve at a given differential pressure and fluid viscosity. In practice for high-viscosity fluids (over 100 cP) and low differential pressures (less than 20 psi), the slip can be neglected.

It is always a good practice to conduct a pump field test soon after the pump has been installed, in order to generate a baseline field-specific curve for future test comparisons.

Shaft Speed

Shaft speed should be measured by a tachometer accompanied with a calibration charts. The measured value should be then recorded, corrected from the charts and finally calculated.

Pressure

Pressures can vary with time so a whole duty cycle should be examined, as the actuation of system valves can cause pressure spikes. The maximum discharge pressure, as well as its average, should be recorded.

Suction Pressure

For the inlet pressure, both maximum and minimum are required in order to establish how well the pump is running. A common mistake is to assume the pressure at the pump inlet flange equal to the tank level pressure and adjust by calculating the losses or by assuming losses are negligible. This approach is particularly wrong for calculating the suction pressure as collapsed filters, debris, solids or other obstructions can make the pressure estimates completely invalid.

The most effective way to measure inlet pressure is with a pressure transducer. Pressure variation generated by a smooth-running pump will mean that pressure gauges need to be glycerin-dampened to prevent them from being damaged.

Inexpensive instruments that display mean and peak values of pressure are available. It is not always possible to fit transducers in ideal positions. However, aiming for the ideal will give more reliable results.

Discharge Pressure

A glycerin-damped calibrated pressure gauge is often sufficient for measuring discharge pressure. If vibrations are a problem (pipe noise, fatigued welds, etc.) then the pressure transducer helps aid the analysis. The discharge pressure usually varies considerably because of varying restrictions to the flow. Absolute accuracy is not usually demanded, so the use of a transducer with a high maximum pressure will avoid the damage from pressure spikes.

Power

The most practical way to measure pump input power is to measure the motor's electric current and voltage. The available shaft power can then be calculated as:

$KW = \frac{I*V*\eta_{motor}}{1000}$ for 1-phase motors

$KW = \frac{I*V*1.732*\eta_{motor}*PF}{1000}$ for 3-phase motors

$BHP = \frac{KW}{0.746}$

The power factor (PF) and motor efficiency can normally be obtained from the motor manufacturer, at least for full load conditions. Both of these items vary widely in different motors and also with changes in load on the same motor.

If there is a gear reducer or any other device coupled between the motor and the pump, that absorbs motor output power, its power losses must be estimated or they can be obtained from the manufacturer. If the pump and motor are directly coupled together, motor output power can be taken as pump input power.

Temperature

In the field, fluid temperature and pump skin temperature, are determined to make sure they are not excessive and thus affecting the pump internal clearances. For certain types of pumps, especially the progressive cavity type, the internal clearances change dramatically with temperature, effecting flow slip, mechanical friction, efficiency and pump life.

Liquid Temperature

Most methods for determining temperature are adequate and ambient temperature usually covers most applications. However anywhere cavitation is being suspected, it is necessary to check the inlet pressure trace at the maximum possible liquid temperature.

Crankcase Temperature

Crankcase temperature is not usually a consideration unless conditions are excessive (over 140° F) and in this case it is necessary to ensure that the oil is capable of performing at this temperature.

Viscosity

Pump flow is virtually independent of liquid viscosity until the viscosity reaches a value where the cylinder does not become filled quickly enough. The maximum recommended rpm and NPIP are available from the pump manufacturer for that specific viscosity value. In case of viscous liquids, the pump should run slower and increased NPIP may be required. Very low viscosity applications are not usually recommended for positive displacement pumps.
Absolute accuracy for determining the viscosity is not needed but the lowest running temperature must be used to determine that value.

1.1.6 Other Tests

Sound Pressure Level

Sound pressure level measurements may be taken at the pump's normal operating condition or may be made in conjunction with one of the performance tests to give a record of sound pressure levels cross a wide range of conditions.
The microphone locations for the test should be approximately 5 feet above the floor or walkway nearest the pump. One reading should be taken from each end of the pump and motor set, approximately 3 feet away from the pump or motor housing.

If the pump and motor are mounted vertically, the reading must be taken at four positions, 90 degrees apart, around the pump. The environment around the pump such as acoustically reflective or absorbent surfaces can have a great influence on the sound pressure measured value. Other equipment operating nearby will also influence sound pressure measurements.

Dynamic testing

Some applications will require a dynamic test before the unit is shipped from the pump manufacturer. Here water or oil would be used in a performance test and a visual check needs to be made of any leakage coming from the seal area. The acceptance criteria may vary depending on the application and normally up to three drops of water per minute or a slight moistening of the exterior surface of the seal chamber with oil is acceptable after the pump has run briefly.

Static testing

When it is possible, the pump containing the seal can be pressurized with a test fluid such as oil or air (air pressure tests are easier to perform) and then blocked off. The pressure decay over a short period of time and is then monitored. Air at 10 psig can be used for testing and units are considered acceptable when the pressure decay is less than 2 psi over a period of 30 seconds.

Dual seal testing

Some applications involving fluids that are hazardous to the environment and present a safety issue to a chemical process plant or refinery may require a dual seal installation in either a double or tandem seal arrangement.
Advances in seal technology and construction materials have allowed positive displacement pumps to be applied to more demanding applications such as hot chemicals and viscous fluids. As heat generated between the seal faces can become significant, manufacturers' recommendations for installation and cooling need to be implemented to avoid premature failure.

1.1.7 Precautions

Safety Precautions

When these pumps are tested, it is important to remember that they move discrete volumes of liquid. These volumes are defined by the displacement of the piston or plunger, are cut off from the inlet feed when the inlet valve closes and are discharged when the cylinder pressure exceeds the pressure on the discharge side.
Therefore it is essential to ensure that there is a safety valve fitted on the discharge side of the pump. If changes are necessary for the tests they must be installed after the safety valve.

Seals

Sealing arrangements for positive displacement pumps are similar to other types of rotating equipment, such as centrifugal pumps. Positive displacement pumps may be sealed with packing or a mechanical seal, depending on the service conditions. The stuffing box or seal chamber is normally located on the suction side of the pump. The seal will then be subject to pump suction pressure or if the pump is running in reverse direction, the seal will be subject to discharge pressure. Some positive displacement pumps, such as screw pumps cannot by design, run in the reverse direction.

Bearings and Rotor Dynamics

The vibration of positive displacement pumps is typically not an issue in regard to expected performance. However, the design of the majority of these pumps can give rise to very large hydraulic forces. As a result of this fact, the pump components must be designed to avoid natural frequencies that could be excited by the system hydraulics.

The rotor-bearing system should be designed so there is essentially rigid shaft performance over the expected operating speed range. This includes the excitation from the once per rev component but, more importantly, the multiple from the number of cylinders or chambers or vanes or gear teeth that produce the hydraulic force frequency to the casing vibration.

Internal (product-lubricated) sleeve journal bearings are often used in many rotary pump designs. The lubrication of these product-lubricating bearings is critical to successful operation.
If viscosity of the pumped fluid is sufficiently high, hydrodynamic lubrication occurs. In this case the rotor does not contact the sleeve and so the composition of the bearing material is not important.
However if the viscosity of the pumped fluid is low, the fluid film supporting the rotor becomes thin or disappears causing the rotor to contact the sleeve. The composition of the bearing material becomes very important in such cases.

1.1.8 Piping/Size considerations

Suction piping

The reducing in cost by using smaller pipe size does compromise sufficient suction pressure in terms of NPSHA at the pump inlet. The smaller the pipe size is, the higher the friction and consequently the greater the NPHA losses are.
Historically and as a rule of thumb, the suction pipe is normally set for flow velocities of 5 ft/sec or less.

Discharge piping

Trying to minimize the installation cost by abruptly changing the size of the discharge pipe can create many problems. The turbulence created by an abrupt change in the discharge piping, known as the *vena contracta* effect; can significantly affect the performance of the pump. The discharge piping is significantly longer than suction piping and thereby has a more significant impact on cost, the rule of thumb and a primary approximation is to keep the velocity in the pump discharge pipe under 15 ft/sec.

Appropriate support pipe

It should be realized that piping issues directly affect the pump's life and its performance. Bringing the pump to the pipe in only one operation and expecting a good pump flange or vessel fit is a very difficult, if not impossible, task. When bringing the pipe to the pump, the last spool should always be left until the pump has been leveled in place and roughly aligned. At this point the pipe should be securely anchored just before the last spool to prevent future movement toward the pump's flanges.

Piping lay out should not be finalized until certified elevation drawings are received from the engineering group and the final isometrics and the piping takeoff can be completed.

1.1.9 Installation

Equipment can be delivered to a site either early or later. When the equipment is late, it is critical to have certified elevation prints of the equipment available when it arrives.
If the equipment is early, it will arrive at the site prior to the construction team's readiness. In such cases, early preparations must be made for long-term storage. It is customary to use oil mist lubrication to keep the equipment in as-shipped conditions during storage. Pressurization of the bearing housing and the casing with 10 to 20 psig H_2O pressure prevents moisture and contaminants from entering the sealed areas and damaging the components.

1.2 Centrifugal Pumps

Two basic components of a centrifugal pump are related to hydraulic performance, the impeller and casing. The radial flow pump is designed to produce a flow pattern through the impeller radially outward and perpendicular to the pump shaft.
A single-stage pump with an end suction and tangential discharge case design illustrated in the figures:

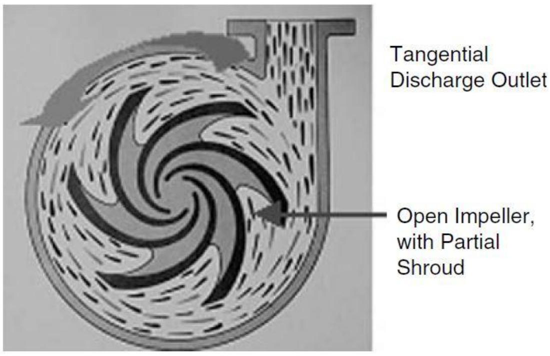


Figure 1.2.a: Open impeller incased by tangential casing

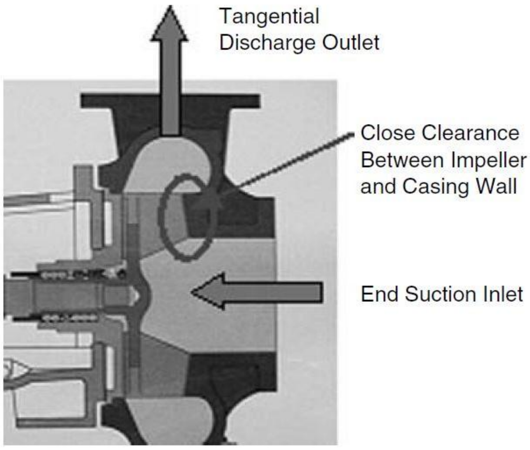


Figure 1.2.b: Tight tolerance casing serves as front of open style impeller

1.2.1 Impeller

Centrifugal pumps are often referred to as kinetic energy machines. Rotation of the impeller causes fluid within the impeller to rotate at a high velocity, imparting kinetic energy to the fluid. This concept is described mathematically by the equation:

$$H_i = \frac{(u_2 * cu_2)}{g}$$

where

H_i= theoretical head developed by the centrifugal pump, in ft
u_2= rotational velocity of the impeller at the outer diameter, in ft/sec
cu_2 = rotational velocity of the fluid as it leaves the impeller, in ft/sec
g = gravitational constant, in ft/sec^2

There are three basic impeller designs:

- A closed impeller that has a shroud (rotating wall) on both the front and the back of the impeller
- A semi-open impeller that has a shroud on one side and is closely fitted to the stationary wall of the casing on the other side
- An open impeller (see Figure 1.2.a) that may or may not have part of a shroud on one side and is closely fitted to the casing wall on the other side (Figure 1.2.b).

As fluid approaches the pump suction, it is assumed to have very little to no rotational velocity. When fluid enters rotating passages of the impeller, it begins to spin at the rotating velocity of the impeller. Fluid is forced outward from the center of the impeller, and its rotating velocity increases in direct proportion to the increasing impeller diameter. The rotating velocity of the impeller can be calculated at any diameter by the equation:

$$u = \frac{D * N}{229}$$

where

u = rotational velocity, in ft/sec
D = diameter at which the velocity is being calculated, in.
N = impeller rotating speed, in rpm
1/229 = constant to convert rpm_in. to ft/sec

The exit velocity of the fluid cu_2approaches the rotating velocity of the impeller u_2 at D_2 but does not equal u_2 in normal operation. The main reason cu_2< u_2 is the ‘‘backward’’ sweep of the impeller vane. The exit velocity of the fluid cu_2 can be calculated from the design parameters of the impeller.

Theoretical head H_i does not account for losses that occur as fluid moves through the impeller during normal operation. Losses in the impeller that normally occur are friction, eddy currents, fluid recirculation, entrance losses and exit losses. Additional losses will occur in the casing. It should be noted that head produced by a centrifugal pump is a function of fluid velocity and is not dependent (normally) on the fluid being pumped. For example, a pump that will produce 100 feet of head on water (8.34 lb/gal) will also produce 100 feet of head on gasoline (6.33 lb/gal).

Fluids with viscosity greater than 20 centipoises will decrease output head produced by a centrifugal pump. For viscous fluids, corrections can be made to predict actual pump performance.

1.2.2 Casing

The function of the pump casing is to:

I. Direct fluid into the eye of the impeller through the suction inlet
II. Minimize fluid recirculation from impeller discharge to impeller suction

III. Capture fluid discharge from the impeller in the case volute to most efficiently utilize work performed by the impeller and direct fluid away from the impeller.

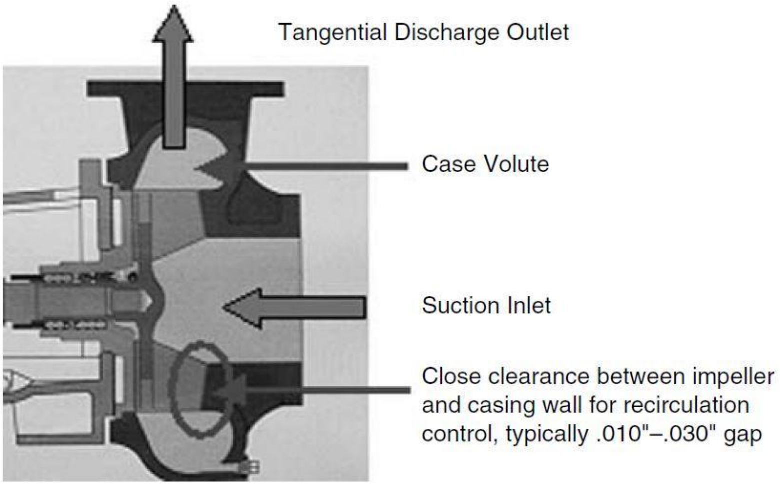


Figure 1.2.2.a: Tight tolerance casing serves as front of open style impeller

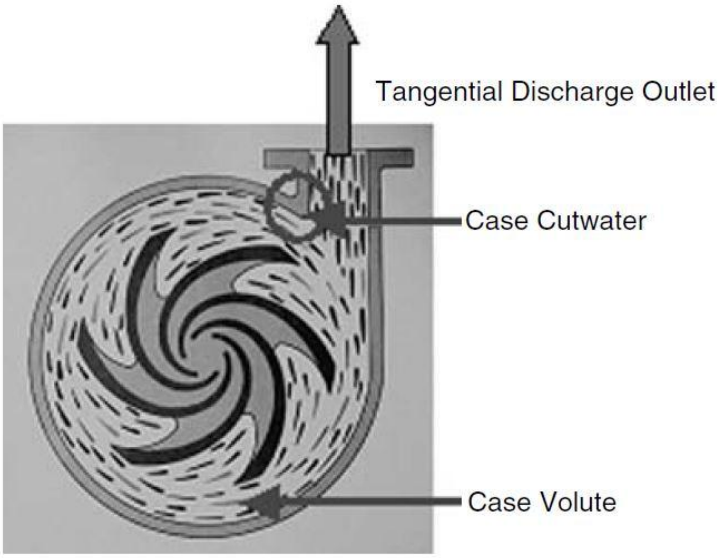


Figure 1.2.2.b: Casing volute and cutwater areas

The impeller performs useful work and increases the head of the fluid. The casing consumes part of the work imparted to the fluid and creates head losses due to friction, eddies and other flow characteristics. A good casing design will minimize the losses, as opposed to a bad casing design.

Typically a centrifugal pump casing is designed so that the suction flange is one or two pipe sizes larger than the discharge flange. This is done to manage velocity of the fluid as it approaches the impeller inlet and also to minimize friction losses ahead of the pump. Excessive losses on the suction side of a pump can cause severe and rapid damage to the pump impeller and casing.
As fluid discharges from the impeller into the case volute, it has increased in head value by the amount of work imparted by the impeller.
Since fluid will naturally flow in the direction of least resistance, it will tend to flow back (recirculate) toward the suction inlet, where fluid entering the impeller is at a relatively low

head. In order to prevent this recirculation, a restriction must be created between the impeller to minimize flow back to the suction and cause the fluid to flow out the discharge. In this example, with an open impeller, that restriction is the very close clearance (gap) between the impeller and the casing wall.

This gap is typically 0.010 to 0.030-inch wide and two major factors determine gap size:

I. Desired performance (the smaller the better)
II. Minimum allowable clearance to prevent impeller rubbing during pump operation.

This works well when the pump is in a new condition but eventually the impeller and casing will begin to wear and the gap size will increase, allowing more fluid to recirculate to the suction side of the impeller. Eventually pump performance will deteriorate to the point that an adjustment to the impeller location will have to be made or the impeller and casing will have to be replaced in order to restore the pump's original performance.

As shown in Figures 1.2.2.a fluid is discharged from the impeller into the case volute. One aspect of a good casing design is that the volume of the volute is sized to match the volume flow through the impeller so that fluid velocity in the volute is somewhat less than fluid velocity exiting the impeller. This reduction in velocity is where a portion of the overall pump head is generated. If velocity reduction is too great, excessive shock losses and eddy currents in the volute will degrade pump head output. If velocity reduction is too small, excessive friction losses will occur in the volute that will also degrade pump head output.

In Figure 1.2.2.b the part of the casing called the cutwater can be seen.
This is where fluid is guided into the discharge outlet and led away from the pump. The cutwater must be accurately located relative to the impeller and angled to minimize flow disruption as fluid exits the casing. A cutwater that is too close to the OD of the impeller will cause a pressure pulse to be created as the impeller vane passes by the cutwater. This momentary high-pressure pulse will disrupt flow and cause pump output to degrade. Conversely, a cutwater that is too far from the impeller will allow too much fluid to pass by the outlet and simply recirculate within the case volute and thereby reduce pump output. The angle of the cutwater must match the flow path of the fluid as it exits the casing or else eddy currents will be created that degrades pump performance.

1.2.3 Concentric Vs Volute Casings

Turbulent flow is detrimental to a centrifugal pump during handling of abrasive fluids. The drilling industry has standardized centrifugal pumps with concentric casings and wide impellers, a design that has proven to offer less turbulence and greatest pump life. The walls of a concentric style of casing (Figure 1.2.3.a) are an equal distance from the impeller throughout the impeller circumference, resulting in a smooth flow pattern.

A volute style of casing (Figure 1.2.3.b) has a cutwater point that disturbs the fluid flow pattern, creating an eddy current.
Wide impellers and larger casing cavities utilized by concentric pumps reduce the effect of the fluid velocity when it exits the impeller OD. This wider area allows fluid to smoothly blend with recirculating fluid within the casing, thus reducing turbulent flow patterns that exist in volute pumps. These characteristics also reduce the sandblasting effect on the ID of the casing that is present with narrow impellers and close-proximity casing walls. Smooth flow patterns and wider recirculation area extend fluid end life and reduces operating costs of the centrifugal pump.

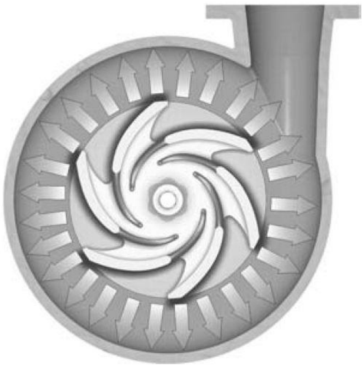


Figure 1.2.3.a.: Concentric style casing

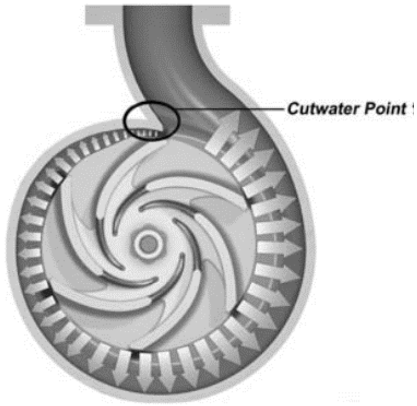


Figure 1.2.3.b: Volute style casing

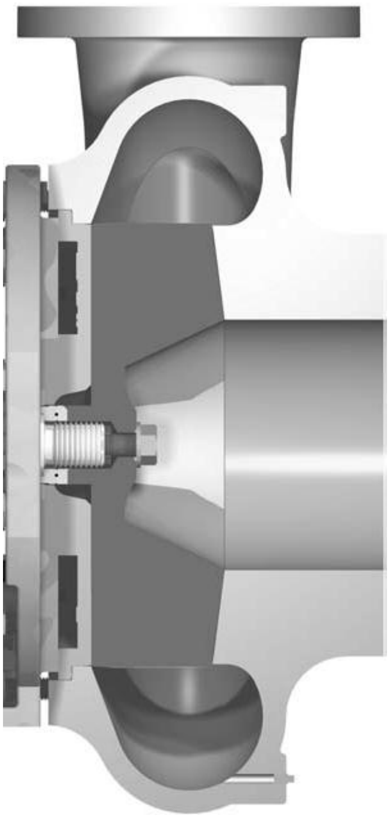


Figure 1.2.3.a: Concentric style casing.

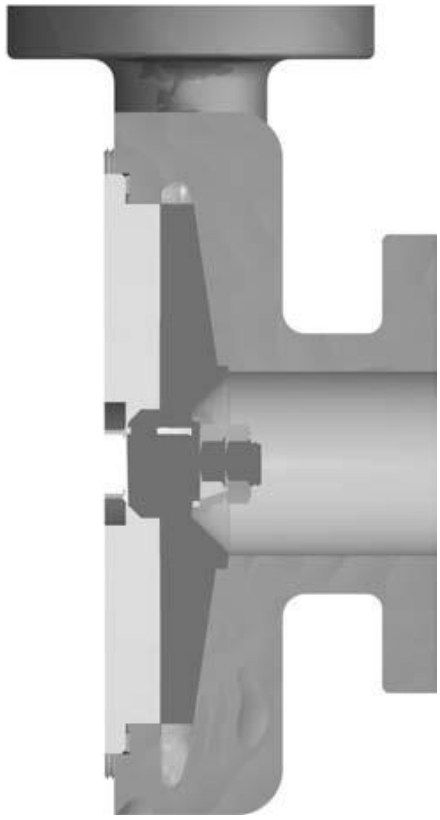


Figure1.2.3.b: Volute style casing.

1.2.4 Centrifugal Pumps Accelerate Fluid

Standard centrifugal pumps are not self-priming and require the fluid end to be primed prior to activation. This can be accomplished by installing the pump in a location that provides a flooded suction or by using a device to prime the pump. Once the pump casing is full of fluid, it can then be energized. Running a pump dry or restricting suction flow can severely damage the fluid end, mechanical seal or packing. The designs of self-priming pumps result in turbulent flow patterns, which cause excessive wear during pumping of abrasive fluids and increase operating costs. The drilling industry avoids using self-priming pumps due to increased downtime and costs.

Once a pump is primed and then activated, suction head at the eye of the impeller drops. Actual positive suction head required at the eye of the impeller to prevent cavitation varies by pump size and flow and is noted on pump curves as NPSHR. When this suction-head drop occurs, atmospheric pressure pushes on the liquid surface and forces it into the pump suction. As fluid enters the pump, the impeller accelerates it. The diameter of the impeller and the rpm at which the impeller is rotated directly affect the velocity of the fluid. The casing of the pump contains this velocity and converts it into head. Casing size and impeller width control the volume that the pump is able to produce.

1.2.5 Cavitation

Consider pump suction located 5 feet above liquid level in the suction tank. Head at the suction flange will be less than atmospheric pressure.

Absolute zero pressure, a perfect vacuum, is 14.7 psi, or 34 feet of water at sea level. If the suction of the centrifugal pump is 35 feet above liquid level in the suction tank, a vacuum will exist and no fluid will enter the pump.

Each centrifugal pump has a minimum suction head required above absolute zero pressure that must exist at the suction to keep the pump full of liquid. A system head that produces a suction pressure less than this value will cause cavitation. If a pump suction line is too small or too long or has too many valves or elbows, the friction loss in the suction line may reduce head to a value below NPSHR.

If the supply tank is at sea level and is vented, inlet pressure to the pump will be:

$$\frac{\text{34 feet vapor pressure in feet}}{SG} \pm liquid\ level\,\frac{above}{below}\,pump\ centerline - suction\ head\ friction\ losses$$

The sum of this calculation is the NPSHA. If NPSHA is greater than NPSHR the pump will function as designed. If NPSHA is equal to or less than NPSHR, the pump will then cavitate. It is advisable to maintain NPSHA at least 3 feet above NPSHR to allow for calculation errors or system changes.

As fluid enters the pump the pressure at the eye of the impeller drops. If insufficient inlet pressure (NPSHA) is present fluid transforms from a liquid state to a gas (boils).

Figure 1.2.5.a: Results of clear fluid cavitation

Gas forms low pressure bubbles and as these bubbles travel from the ID to the OD of the impeller pressure increases. Eventually the pressure increases enough to collapse the low pressure bubbles. When this occurs the bubbles implode and space once occupied by the bubbles fills with fluid. Fluid fills this space with such force that it actually fractures adjacent metal. As this process repeats it will knock out sections of the fluid end and can even knock a hole through the stuffing box, impeller, or casing. Cavitation can be caused by improper suction or discharge conditions and can severely reduce the life of pump.
Figures 1.2.5.a and 1.2.5.b show the result of severe cavitation in a pump that was transferring clear fluid. Notice that the fractured metal has sharp corners.
Figure 1.2.5.c shows the result of cavitation in a pump that was handling abrasive fluids. After the fracture occurs sharp corners are worn smooth by abrasive fluid. The damaged part will look as if a spoon were used to scoop sections of metal from the part. When a pump is severely cavitating it sounds like it is pumping gravel.

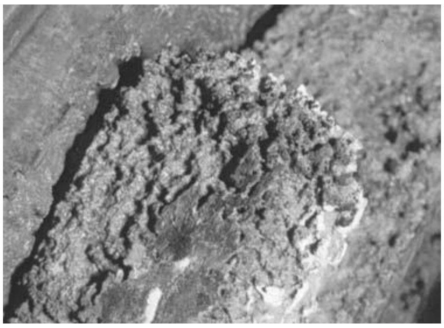


Figure 1.2.5.b: Results of clear fluid cavitation

Figure 1.2.5.c: Results of abrasive fluid cavitation

Entrained Air

Entrained air in transferred fluid causes excessive turbulent flow patterns and can vapor lock the pump. Air bubbles do not collapse like low pressure bubbles during cavitation.
As an air bubble enters the pump, it moves from the ID toward the OD of the impeller. Increased pressure at the OD of the impeller pushes the bubble back into the ID, where it combines with other air bubbles to become a larger bubble. This process continues to occur until the bubble at the ID of the impeller becomes large enough to impede suction flow which can cause cavitation or it becomes as large as the suction inlet and prevents fluid from entering the pump, resulting in vapor lock. Once the pump is shut down, the bubble will normally escape through the discharge, but when the pump is restarted, the process repeats itself. Recirculation of air bubbles from the ID to the OD and back again also causes turbulent fluid flow patterns that will result in excessive pump wear. Air entrained in fluid, can enter a pump through a loose flanged or threaded connection, through the pump packing when the pump has a high lift requirement or through an air vortex formed in the suction tank.

1.2.6 Sizing Centrifugal Pumps

Centrifugal pumps are available in a variety of materials, configurations, sizes and designs. Normally, a single size configuration and speed can be selected to best meet the intended application. Accurate centrifugal pump selection can occur only with knowledge of system details. It is imperative to obtain accurate information such as fluid temperature, specific gravity, pipe diameter, length of pipe, fittings, elevations, flow required, head required at end of transfer line, type of driver required and type of power available. Without all of this information, assumptions have to be made that could cause pump failure, high maintenance costs, downtime and improper performance.

Standard Definitions-Terms associated with centrifugal pumps

Flow rate

It is the volume of liquid going through a pipe in a given time. If a hose stream will fill a 10-gal bucket in 2 minutes, then the flow rate is 5 gpm. The bottom axis of most pump curves is flow rate and is usually measured in gallons per minute or cubic meters per hour ($^{m^3}/_{hr}$).

Friction loss

Resistance to movement of fluid within the pipe or head loss caused by turbulence and dragging that results when fluid comes into contact with the ID of pipe, valves, fittings, etc. This value is normally measured in feet per 100 feet of pipe and is noted in Table 1.2.6.a under the column heading ‘‘Friction Loss in Feet Head per 100 Ft of Pipe.’’ These tables are based on Schedule (SCH)-40 new steel pipes. Other piping material or older scaled or pitted pipe will have higher friction losses.

Head

Distance in feet that water will rise in an open-ended tube connected to the place where the measurement is to be taken. Units of psi vary with the weight of the fluid, but head in feet or meters is constant regardless of fluid weight.

Net positive suction head available (NPSHA)

It is the amount of head that will exist at the suction flange of the pump above absolute zero. Friction losses, atmospheric pressure, fluid temperature/vapor pressure, elevation and specific gravity affect this value and it must be calculated.

Net positive suction head required (NPSHR)

Amount of inlet head above absolute zero required by the centrifugal pump to operate properly. This value varies with pump size and flow rate and is normally represented on the pump curve.

Total differential head (TDH)

Amount of head produced by a centrifugal pump in excess of pump inlet head. This value is found on the left axis of most pump curves.

Total discharge head

Sum of the inlet head and total differential head of a centrifugal pump, measured in feet or meters.

Velocity (v ft/sec)

Refers to the average speed that liquid travels with through a pipe. Velocity is measured in feet per second (ft/sec). Velocities measured in ft/sec can be found in Tables 1.2.6.a, where one can look up the velocity for recommended flow rates in a pipe size or the flow rate in a pipe for any velocity desired. For example 10 ft/sec in an SCH 40 4-inch-diameter pipe will flow 400 gpm.

6-Inch Nominal	Steel Schedule-40 Pipe ID: 6.065 Inches €/D: 0.000293			8-Inch Nominal	Steel Schedule-40 Pipe ID: 7.981 Inches €/D: 0.000226		
Flow Rate (gpm)	v (ft/sec)	$v^2/2g$ (ft)	Friction Loss in Feet Head per 100 Ft of Pipe	Flow Rate (gpm)	v (ft/sec)	$v^2/2g$ (ft)	Friction Loss in Feet Head per 100 Ft of Pipe
100	1.11	0.02	0.08	60	1.03	0.02	0.05
120	1.33	0.03	0.12	180	1.15	0.02	0.06
140	1.55	0.04	0.16	200	1.28	0.03	0.08
160	1.78	0.05	0.20	220	1.41	0.03	0.09
180	2.00	0.06	0.25	240	1.54	0.04	0.11
200	2.22	0.08	0.30	260	1.67	0.04	0.13
220	2.44	0.09	0.36	280	1.80	0.05	0.14
240	2.66	0.11	0.42	300	1.92	0.06	0.16
260	2.89	0.13	0.49	320	2.05	0.07	0.18
280	3.11	0.15	0.56	340	2.18	0.07	0.21
300	3.33	0.17	0.64	360	2.31	0.08	0.23
320	3.55	0.20	0.72	380	2.44	0.09	0.25
340	3.78	0.22	0.81	400	2.57	0.10	0.28
360	4.00	0.24	0.90	450	2.89	0.13	0.35
380	4.22	0.28	1.00	500	3.21	0.16	0.42
400	4.44	0.31	1.10	550	3.53	0.19	0.51
420	4.66	0.34	1.20	600	3.85	0.23	0.60
440	4.89	0.37	1.31	650	4.17	0.27	0.70
460	**5.11**	**0.41**	**1.42**	700	4.49	0.31	0.80
480	**5.33**	**0.44**	**1.54**	750	4.81	0.36	0.91
500	**5.55**	**0.48**	**1.66**	**800**	**5.13**	**0.41**	**1.02**
550	**6.11**	**0.58**	**1.99**	**850**	**5.45**	**0.46**	**1.15**
600	**6.66**	**0.69**	**2.34**	**900**	**5.77**	**0.52**	**1.27**
650	**7.22**	**0.81**	**2.73**	**950**	**6.09**	**0.58**	**1.41**
700	**7.77**	**0.94**	**3.13**	**1000**	**6.41**	**0.64**	**1.56**
750	**8.33**	**1.08**	**3.57**	**1100**	**7.05**	**0.77**	**1.87**
800	**8.88**	**1.23**	**4.03**	**1200**	**7.70**	**0.92**	**2.20**
850	**9.44**	**1.38**	**4.53**	**1300**	**8.34**	**1.08**	**2.56**
900	**9.99**	**1.55**	**5.05**	**1400**	**8.98**	**1.25**	**2.95**
950	**10.5**	**1.73**	**5.60**	**1500**	**9.62**	**1.44**	**3.37**
1000	**11.1**	**1.92**	**6.17**	**1600**	**10.3**	**1.64**	**3.82**
1100	**12.2**	**2.32**	**7.41**	**1700**	**10.9**	**1.85**	**4.29**
1200	13.3	2.76	8.76	**1800**	**11.5**	**2.07**	**4.79**
1300	14.4	3.24	10.2	**1900**	**12.2**	**2.31**	**5.31**
1400	15.5	3.76	11.8	2000	12.8	2.56	5.86
1500	16.7	4.31	12.5	2200	14.1	3.09	7.02
1600	17.8	4.91	15.4	2400	15.4	3.68	8.31
1700	18.9	5.54	17.3	2600	16.7	4.32	9.70
1800	20.0	6.21	19.4	2800	18.0	5.01	11.20
1900	21.1	6.92	21.6	3000	19.2	5.75	12.8
2000	22.2	7.67	23.8	3200	20.5	6.55	14.5

Table 1.2.6.a

Head Produces Flow

Most water used at home and in industry comes from tank towers or standpipes. Figure 1.2.6.b shows water flowing through a straight pipe of constant diameter lying on level ground. If a clear sight tube is installed on the pipe near the open end, it can be used to measure head at that place. By closing the end of the pipe, flow will stop and the water in the sight tube will rise to a level equal to that of the standpipe. When the end of the pipe is opened fully, the flow will be the most the standpipe head can deliver, and the water in the sight tube will drop to the bottom. Almost all of the head is consumed while pushing the water through the pipe and overcoming friction.

The velocity head is used first, to speed the water from a standstill in the standpipe up to the velocity in the pipe as it enters. The velocity head (in ft) depends only on the velocity of the flow (in ft/sec), not on diameter or gpm. It is the same amount at any point in the pipe (constant diameter) even to the open end, where it shows as the strength of the flow stream. It is usually small, 3–6% of the total head for pipes of 100 feet or more in length. It is shown in the friction loss tables Table 1.2.6.a under the column headed ‘‘V^2/2g.’’

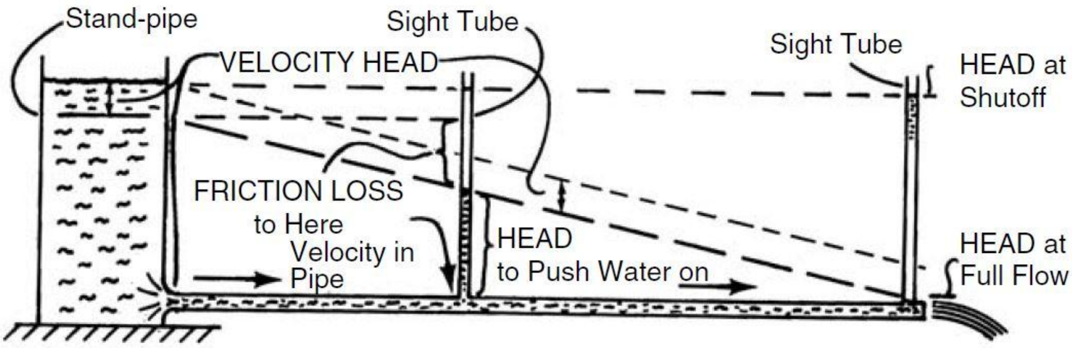


Figure 1.2.6.b: Head pressure measured by a standpipe

A sight tube installed somewhere near the halfway point will show the (pressure) head remaining at that point that pushes the water on to the end. The difference of the height in the sight tube from the height at shutoff is the velocity head plus the friction loss from the standpipe to that point.

The use of a standpipe to supply fluid for pipe friction problems is the clearest way to demonstrate how pipe friction tables are made. While a standpipe illustration explains the system head and flow, it is not a practical method of producing head in most applications and pumps are substituted for standpipes.
Pumps can be sized to produce the proper amount of head to achieve the desired flow rate and overcome the friction losses and elevation in a system.

1.2.7 Read Pump Curves

A centrifugal pump curve comprises a grid depicting head and flow rate and a series of lines that illustrate pump performance characteristics. Figure 1.2.7.a is a typical pump curve, and each set of lines will be reviewed individually.
Figure 1.2.7.a is a curve for a Magnum pump, size 8 (suction) x 6(discharge) x 14 (maximum impeller size) inches, operating at 1750 rpm.
(Note: The suction will always be equal to or greater than the pump discharge size.)

Impeller sizes, from 10 to 14 inches in diameter, can pass a spherical solid up to 1 3/8 inches in diameter. All this information is located above the grid. Performance characteristics of this pump will change if the speed is altered and this curve simply shows performance when the pump is driven at 1750 rpm.

Figure 1.2.7.a utilizes a left and bottom axis. Other curves may include a top and right axis. The left axis denotes the scale for TDH. This is the amount of head in feet; the pump will produce in excess of suction head. The bottom axis denotes the scale for capacity gpm. In order to read a curve review each set of lines individually. In Figure 1.2.7.b, several points have been marked on a line, which all depict a flow rate of 1500 gpm. If a flow rate of 1200 gpm were desired, it would be necessary to estimate the position on the grid.

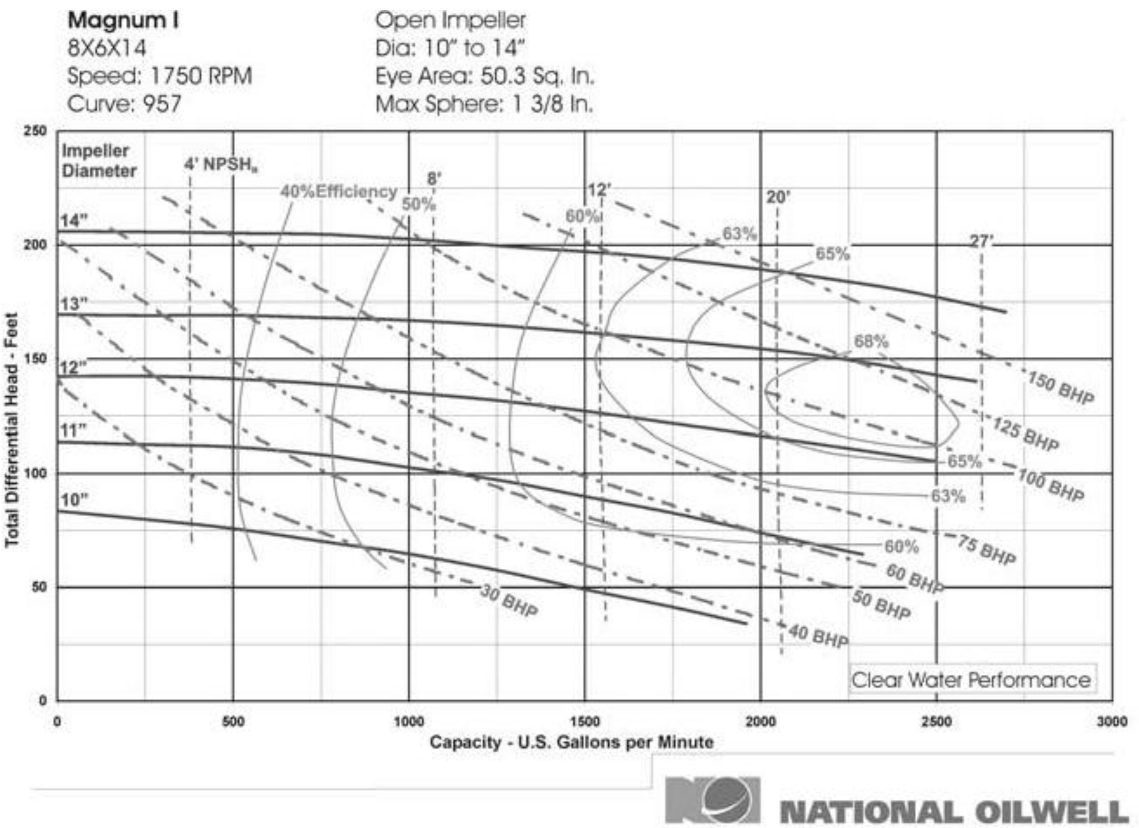


Figure 1.2.7.a: Centrifugal pump curve

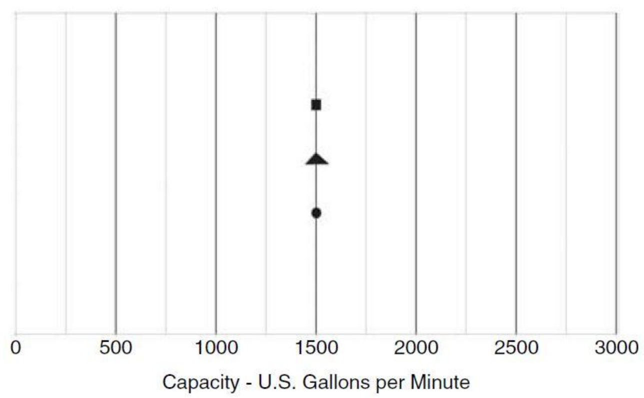


Figure 1.2.7.b: Designation of 1500 GPM operating points

Figure 1.2.7.c shows several points marked on the line that all depict a TDH of 100 feet. If the pump had an inlet suction head of 20 feet and were sized to produce 100 feet of head, then the

pump discharge head (total discharge head) would be equal to 120 feet. If TDH were 100 feet and fluid had to be lifted 5 feet above the suction liquid level, the total discharge head would be 95 feet.

Suppose that 8-inch-diameter SCH 40 new pipe will be used and that it lies on level ground. Table 1.2.6.a shows that at 1500 gpm, friction loss per 100 feet of pipe will be 3.37 feet. Pipe 3000 feet long will have 30 times (3000/100) as much friction loss, or 101 feet:

(30)(3.37) = 101

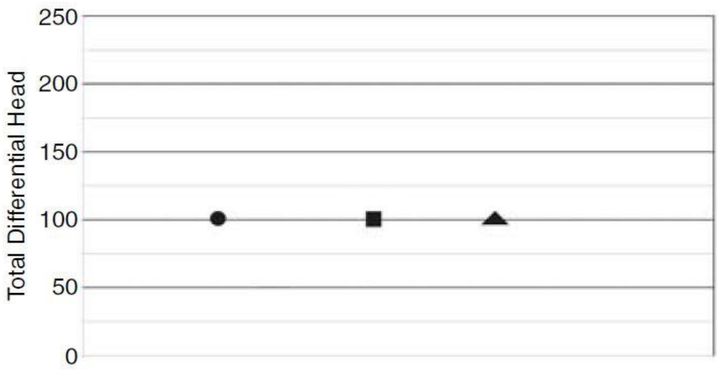


Figure 1.2.7.c: Designation of 100 feet total differential head

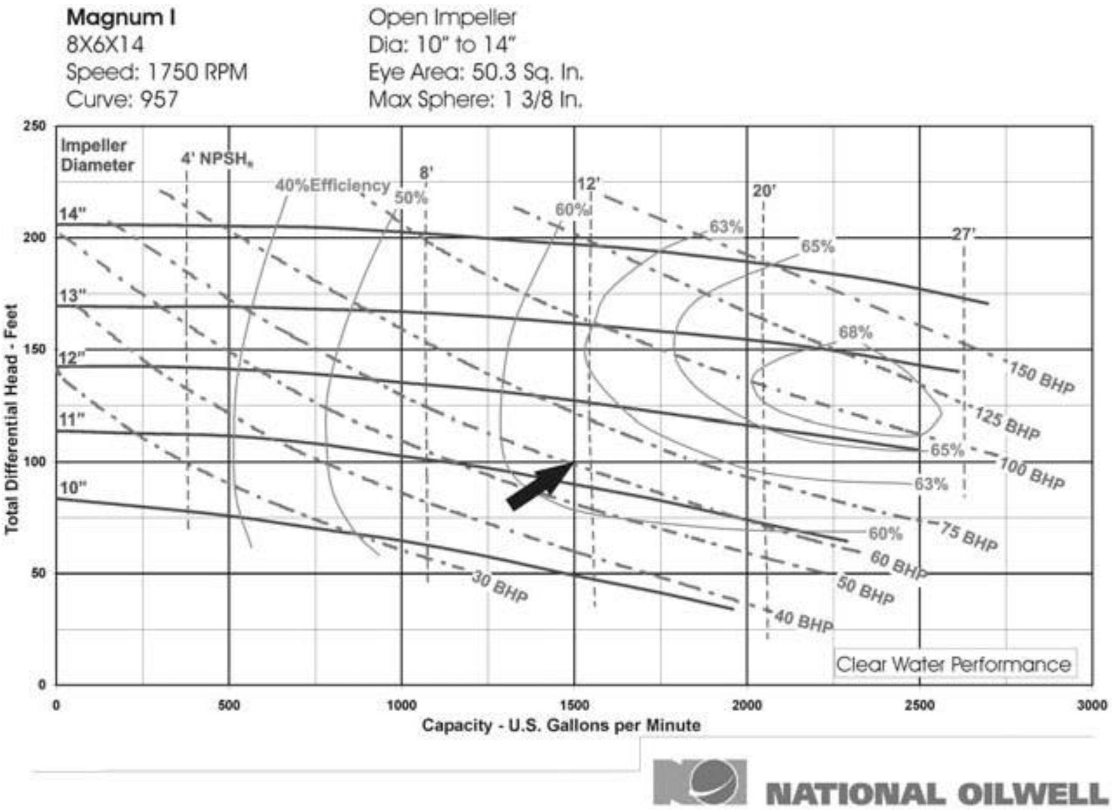


Figure 1.2.7.d: Impeller diameter, HP, efficiency and NPSHR lines

Therefore 8-inch pipe 3000 feet long will require 101 feet of head to flow 1500 gpm. A curve can be marked at 1500 gpm at 101 feet of head.

The solid curved lines in Figure 1.2.7.d that extend from the left axis to the right and are labeled 10'', 11'', 12'', etc., designate the diameter of the impeller, the TDH produced by the impeller at this speed and the flow rate produced by the impeller. The pump discharge head can be altered by changing rpm or by changing impeller diameter. If a fixed-speed driver is utilized, such as 1750 rpm, the only way to vary pump head is to alter the impeller diameter.

If the requirement is for 1500 gpm at 101 feet of head and the operating speed is 1750 rpm, the impeller would have to be trimmed to 11 $\frac{1}{4}$ inches. Impellers can be sized to $\frac{1}{8}$ inch increments, but for common installations $\frac{1}{4}$ inch increment is sufficient.

The dash-dot curves in Figure 1.2.7.d running diagonally from upper left to lower right labeled 30, 40, 50 BHP (brake horsepower), etc., designate hp required to transfer clear water. In this example the pump requires 60 hp for clear water. This value must be corrected for fluids with an SG other than 1.0.

Solid curves running from top to bottom in a circular pattern and labeled 40, 50, 60%, etc., designate the pump efficiency. The higher the efficiency level the lower the power operating cost. Concentric casing pumps have lower efficiency levels than other styles of pump. However concentric-style pumps last longer and have less downtime and maintenance operating costs when transferring abrasive fluids. In the preceding example the pump is approximately 61% efficient.

Dashed lines running from top to bottom designate minimum NPSHR for the pump to operate properly. In this example the pump has an NPSHR of approximately 11–12 feet.

Friction Loss Tables

When selecting pipe size, both friction losses and line velocity must be considered. Friction losses should be kept to a minimum in order to reduce hp requirements. For that reason the discharge piping should be sized to achieve a flow rate of 5 to 12 1/2 ft/sec and suction piping should be sized to achieve a flow rate of 5 to 8 1/2 ft/sec. A minimum line velocity of 5 ft/sec is recommended to avoid solids within the liquid to settle in the piping and thereafter it will be difficult to open and close valves.

Exceeding 12 1/2 ft/sec on the discharge line will cause excessive wear of valves, elbows, and tees.

Exceeding 8 1/2 ft/sec on the suction line will cause excessive wear of the pump fluid end due to turbulent flow patterns that will occur as the fluid impacts the impeller.

Table 1.2.7.e provides line velocities and friction losses that occur when fluid travels through new SCH 40 steel pipe. Scaled pipe and piping with different IDs will have different values and corrections to shown values must be made. A design factor of 15–20% minimum should be added to values for this table. The values in boldface designate recommended optimum flow rates based on discharge line velocities.

To determine line velocities and friction losses first locate the proper pipe diameter. In the ‘‘Flow Rate (gpm)’’ column, locate the maximum anticipated flow rate. Line velocities will be located in the ‘‘V (ft/sec)’’ column. Friction loss values will be located in the column ‘‘Friction Loss in Feet of Head per 100 Ft of Pipe.’’ For example in Table 1.2.6.a for a 6-inch nominal pipe size at 1000 gpm, line velocities are 11.1 ft/sec and friction losses per 100 feet of pipe will be 6.17 feet of head.

Friction Loss and Elevation Considerations

Once the head requirement at the end of the transfer line and volume required is known, it is time to determine elevation and friction losses that need to be overcome.

Elevation is the distance above or below the centerline of the pump. Therefore, if a centrifugal is mounted on deck 1 and the transfer line ends on deck 2 which is 20 feet above the pump centerline, then the discharge elevation is 20 feet. These 20 feet of elevation must be added to the discharge head required.

Additionally, the suction supply tank elevation must be considered. If minimum liquid surface is 8 feet above the pump centerline, then 8 feet of positive head will feed the pump and can be subtracted from the discharge head requirement.

However, if the minimum liquid surface level is 8 feet below the pump centerline, a negative head is created and this must be added to the discharge head requirement.
The friction loss varies with flow rate and pipe diameter. Take, for example, a contractor who wishes to operate a two-cone desander equipped with 10-inch cones and anticipates the maximum mud weight to be 16 lb/gal. The pump is mounted on the same deck as the desander and is 150 feet away. The inlet to the desander is 10 feet above the deck. The supply tank minimum liquid surface level is 8 feet above the pump centerline. Remember that two 10-inch cones will flow 1000 gpm and that the desander requires 80 feet of inlet head:

$$\begin{aligned}80\ feet\ required\ by\ desander + 10\ feet\ of\ discharge\ elevation \\ - 8\ feet\ of\ positive\ suction + Discharge\ friction\ loss \\ + \ Suction\ friction\ loss \\ = TDH\ required\ at\ discharge\ of\ centrifugal\ pump\end{aligned}$$

Line Size Schedule-40 Steel Pipe	Velocity (ft/sec)	Head Loss (ft/100 ft)
4	25.5	64.8
5	16	15.8
6	11.1	6.17
8	6.41	1.56
10	4.07	0.50

Chart based on 1000-gpm flow rate.

Table 1.2.7.e: Known friction losses

At 1000 gpm, 4- and 5-inch lines have line velocities that exceed the maximums recommended and should therefore not be used. A 10-inch line has a velocity that does not meet minimum velocity requirements and settling may occur. Line velocities for 6-inch pipe are excessive for the suction side of the pump. This means that optimum-size suction piping for this application is 8 inches. Discharge piping could use either 6-inch or 8-inch pipe but it would be most economical to utilize 6-inch piping.

Since suction and discharge line sizes have been selected, friction losses can be calculated. In this example assume that the discharge line is new 6-inch SCH 40 and has one butterfly valve, six ells, one running tee and one branched tee and is 150 feet long.
The suction line is new 8-inch SCH 40 and has one elbow, one branched tee and one butterfly valve and is 30 feet long. Each fitting cause friction losses that can be measured and compared to equivalent feet of pipe (see Table 1.2.7.f)
Discharge line:

- (1) 6-inch butterfly valve = 22.7 feet
- (6) 6-inch ells = 91.2 feet
- (1) 6-inch running tee = 10.1 feet
- (1) 6-inch branched tee = 30.3 feet

Actual feet of pipe = 150 feet. Total = 304.3 equivalent feet of 6-inch pipe.

Suction line:

- (1) 8-inch ells = 20 feet
- (1) 8-inch branched tee = 39.9 feet
- (1) 8-inch butterfly valve = 29.9 feet

Actual feet of pipe = 30 feet. Total = 119.8 equivalent feet of 8-inch pipe. Now calculate friction losses using Table 1.2.7.f. Friction loss values are based per 100 feet of pipe. Divide the equivalent feet of pipe by 100 to determine the multiplier. Table 1.2.7.e is based on new steel pipe, and even if new steel pipe is utilized, a 20% design factor should be added to the friction loss values. For pipe other than new SCH 40, refer to engineering handbooks for friction loss values. Three hundred four equivalent feet of 6-inch discharge line flowing 1000 gpm would have a friction loss of 18.76 feet (3.04 feet x 6.17 = 18.76). With a 20% design factor, the value is 22.5. One hundred twenty equivalent feet of 8-inch suction line flowing 1000 gpm would have a friction loss of 1.87 feet (1.20 x 1.56 = 1.87). With a 20% design factor, the value is 2.24.

Discharge elevation above pump centerline is 10 feet; supply tank liquid surface level is 8 feet above the pump centerline; and the required desander inlet head is 80 feet. Therefore, 80+10-8+22.5+2.24 = 106.74 TDH required.

Knowing the flow rate of 1000 gpm and TDH required at the pump to be 107 feet, an individual can begin the pump selection process.

Centrifugals must be sized using maximum values anticipated, to ensure that they can perform without cavitation and that the driver is adequately sized. If there is a possibility that the contractor would add a third cone and flow at 1500 gpm, the pump must be sized to handle up to 1500 gpm (this would also affect line velocities and friction losses).

Motors must be sized for maximum mud weight. For this example, assume 1000 gpm to be the maximum flow rate and 16 lb/gal mud to be maximum mud weight. A pump to produce 1000 gpm at 107 feet TDH is required.

Friction Loss in Pipe Fittings in Terms of Equivalent Feet of Straight Pipe

Nominal Pipe Size	Actual Inside Diameter	Gate Valve (f.o.)	90° Elbow	Long Radius 90° or 45° std. elbow	Std. Tee (thru flow)	Std. Tee (branch flow)	Close Return Bend	Swing Check Valve (f.o.)	Angle Valve (f.o.)	Globe Valve (f.o.)	Butterfly Valve
$1\frac{1}{2}$	1.61	1.07	4.03	2.15	2.68	8.05	6.71	13.4	20.1		
2	2.067	1.38	5.17	2.76	3.45	10.3	8.61	17.2	25.8	7.75	7.75
$2\frac{1}{2}$	2.469	1.65	6.17	3.29	4.12	12.3	10.3	20.6	30.9	9.26	9.26
3	3.068	2.04	7.67	4.09	5.11	15.3	12.8	25.5	38.4	11.5	11.5
4	4.026	2.68	10.1	5.37	6.71	20.1	16.8	33.6	50.3	15.1	15.1
5	5.047	3.36	12.6	6.73	8.41	25.2	21	42.1	63.1	18.9	18.9
6	6.065	4.04	15.2	8.09	10.1	30.3	25.3	50.5	75.8	22.7	22.7
8	7.981	5.32	20	10.6	13.3	39.9	33.3	58	99.8	29.9	29.9
10	10.02	6.68	25.1	13.4	16.7	50.1	41.8	65	125	29.2	29.2
12	11.938	7.96	29.8	15.9	19.9	59.7	49.7	72	149	34.8	34.8
14	13.124	8.75	32.8	17.5	21.8	65.6	54.7	90	164	38.3	38.3
16	15	10	37.5	20	25	75	62.5	101	188	31.3	31.3
18	16.876	16.9	42.2	22.5	28.1	84.4	70.3	120	210	35.2	35.2
20	18.814	12.5	47	25.1	31.4	94.1	78.4	132	235	39.2	39.2

f.o. = full open.
Calculated from data in Crane Co., Technical Paper 410.

Table 1.2.7.f

Following are possible pump selections for this application:

5 × 4 × 14	11.50″ impeller	39 hp (water)	29′ $NPSH_R$	70% efficiency
6 × 5 × 11	10.75″ imp	39 hp (water)	9′ $NPSH_R$	71% eff.
6 × 5 × 14	10.75″ imp	39 hp (water)	10′ $NPSH_R$	70% eff.
8 × 6 × 11	10.75″ imp	40 hp (water)	10′ $NPSH_R$	67% eff.
8 × 6 × 14	11.25″ imp	50 hp (water)	7′ $NPSH_R$	54% eff.
10 × 8 × 14	12.25″ imp	71 hp (water)	16′ $NPSH_R$	40% eff.

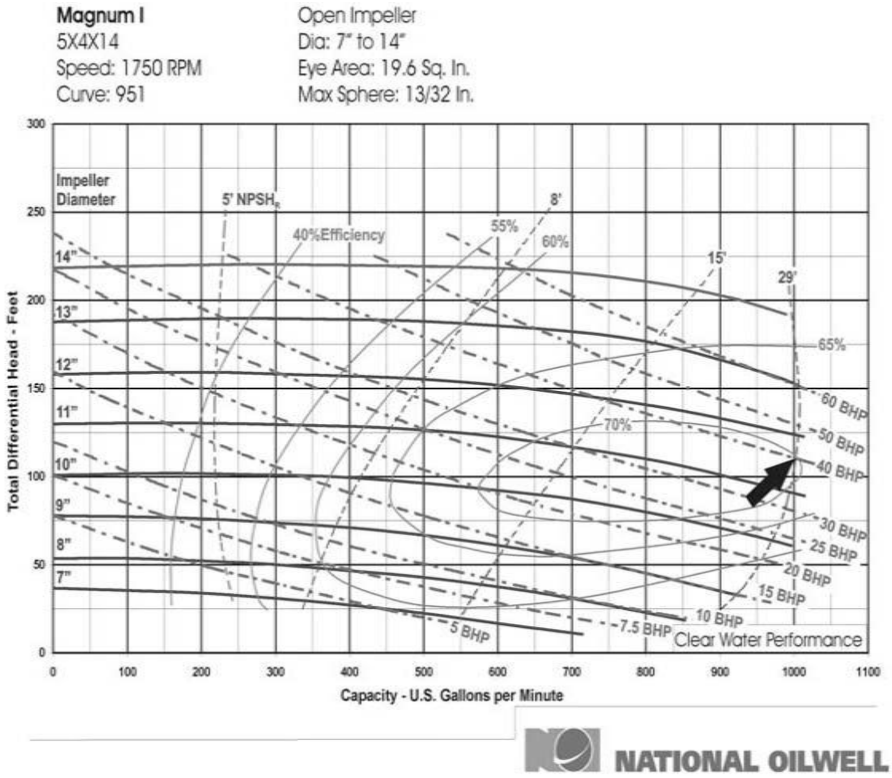


Figure: 5 x 4 x14

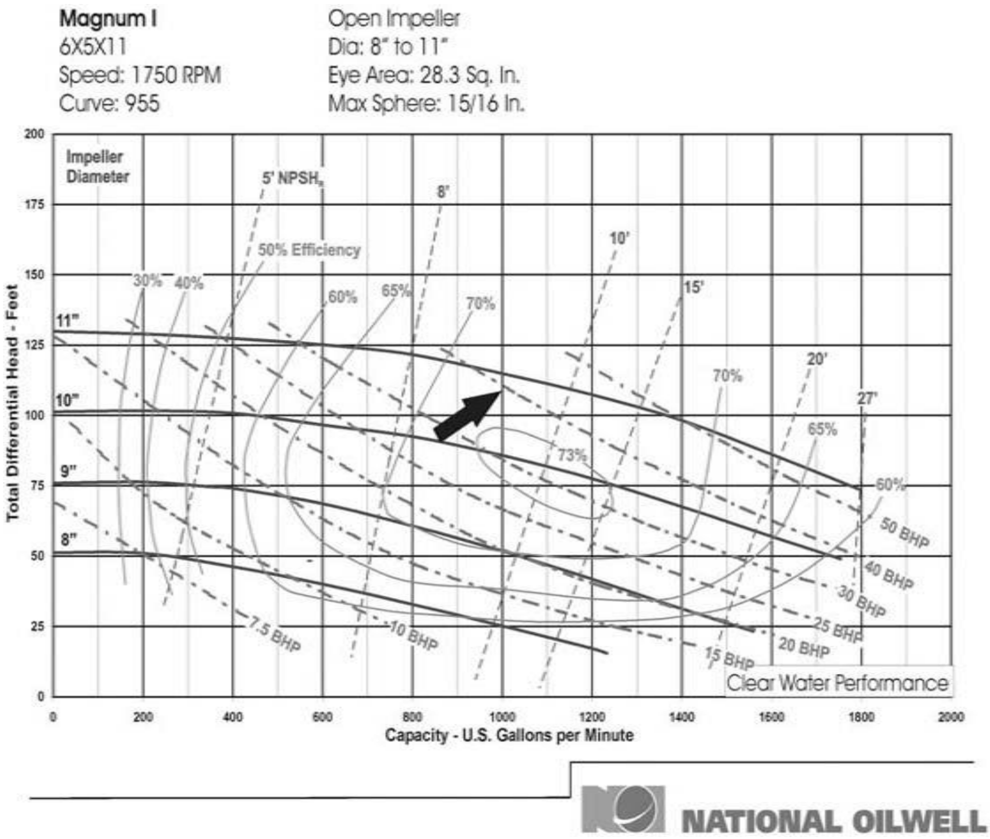


Figure: 6 x 5 x 11

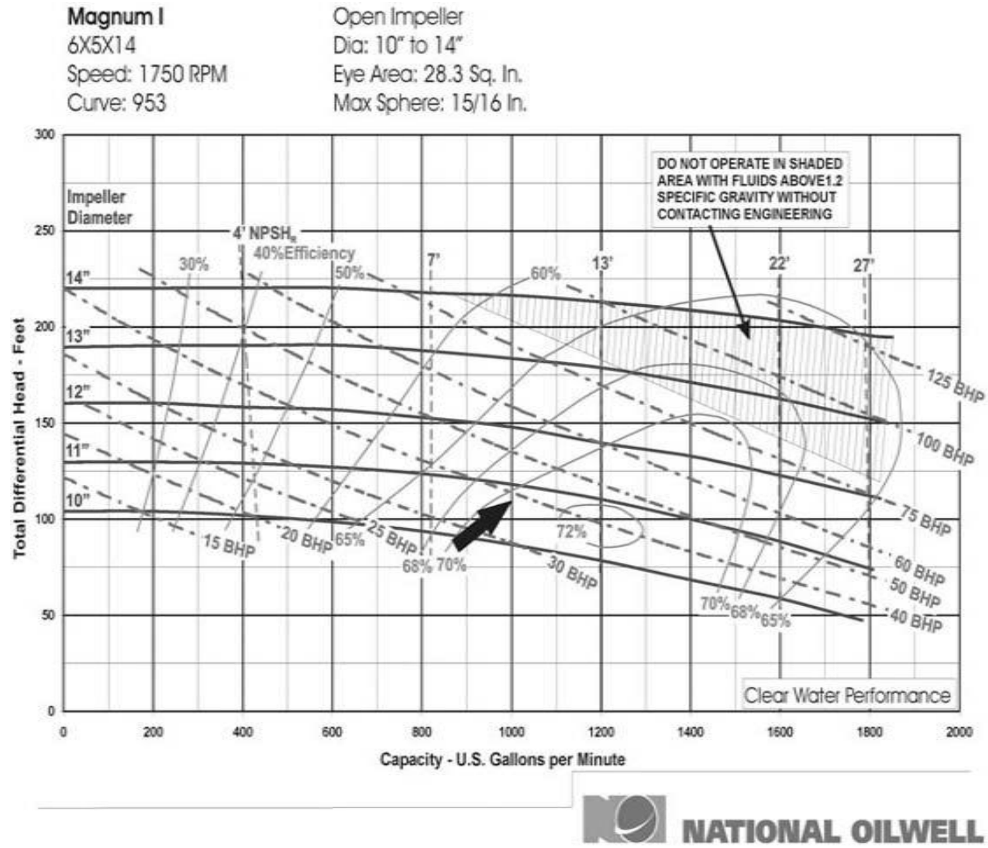


Figure: 6 x 5 x 14

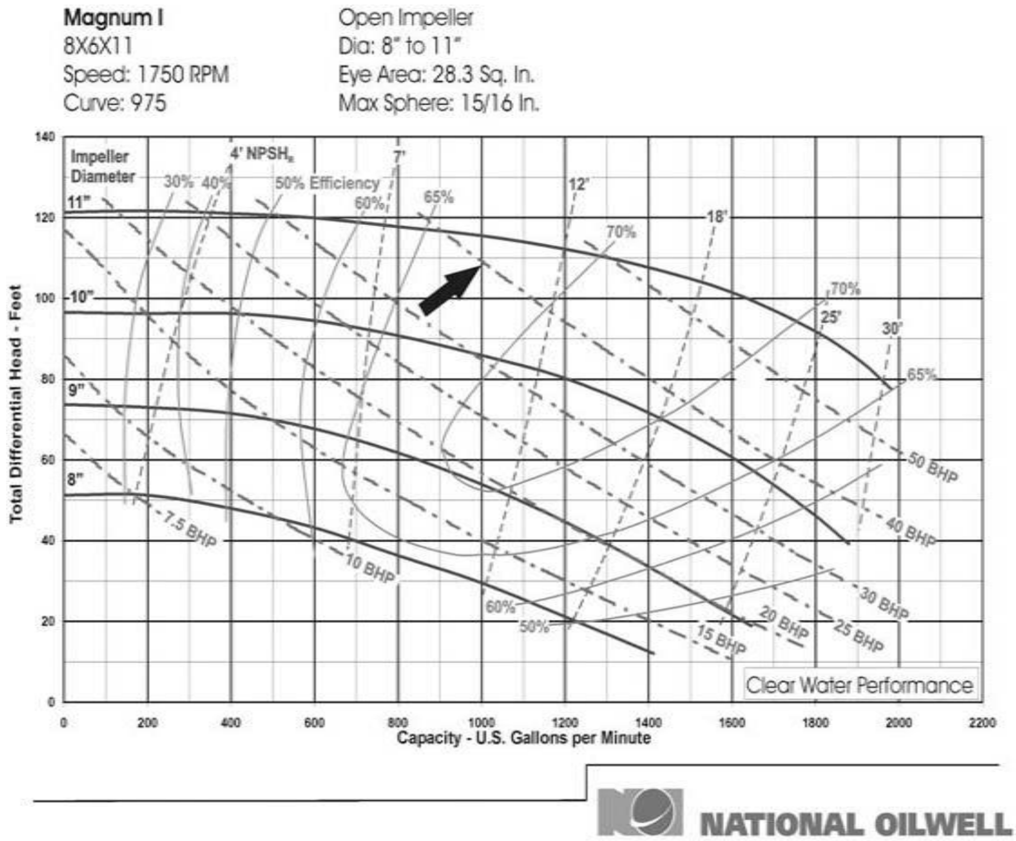


Figure: 8 x 6 x 11

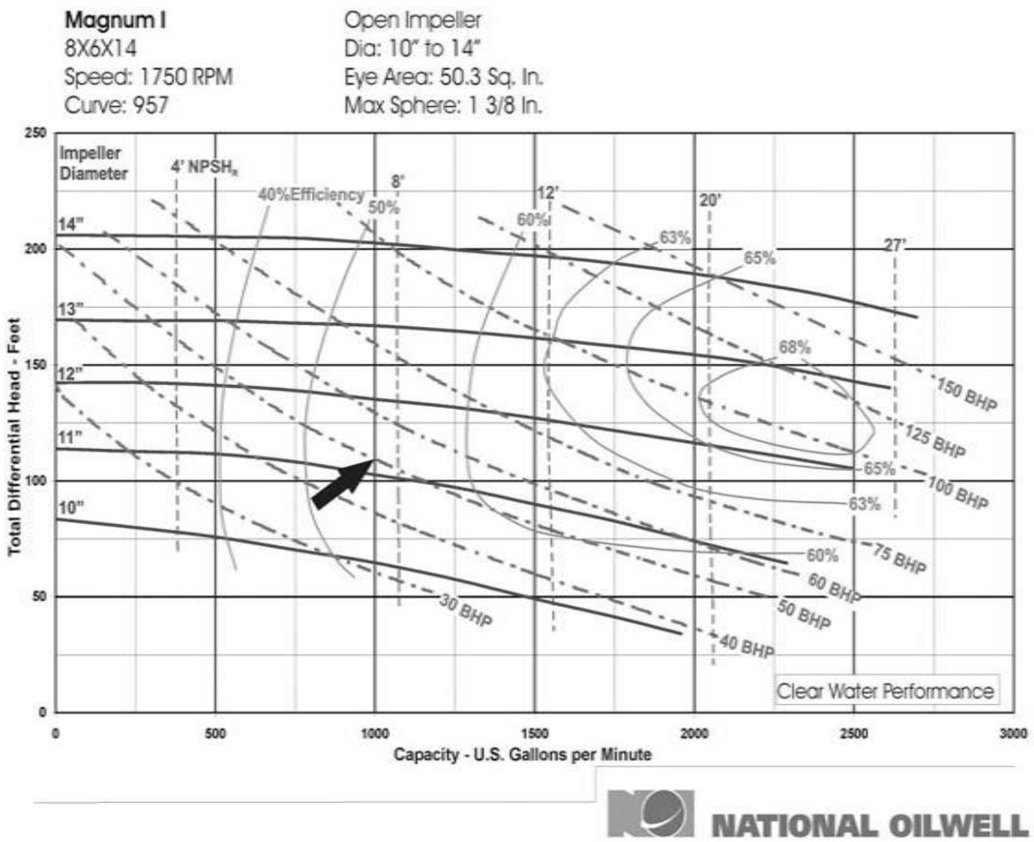


Figure: 8 x 6 x 14

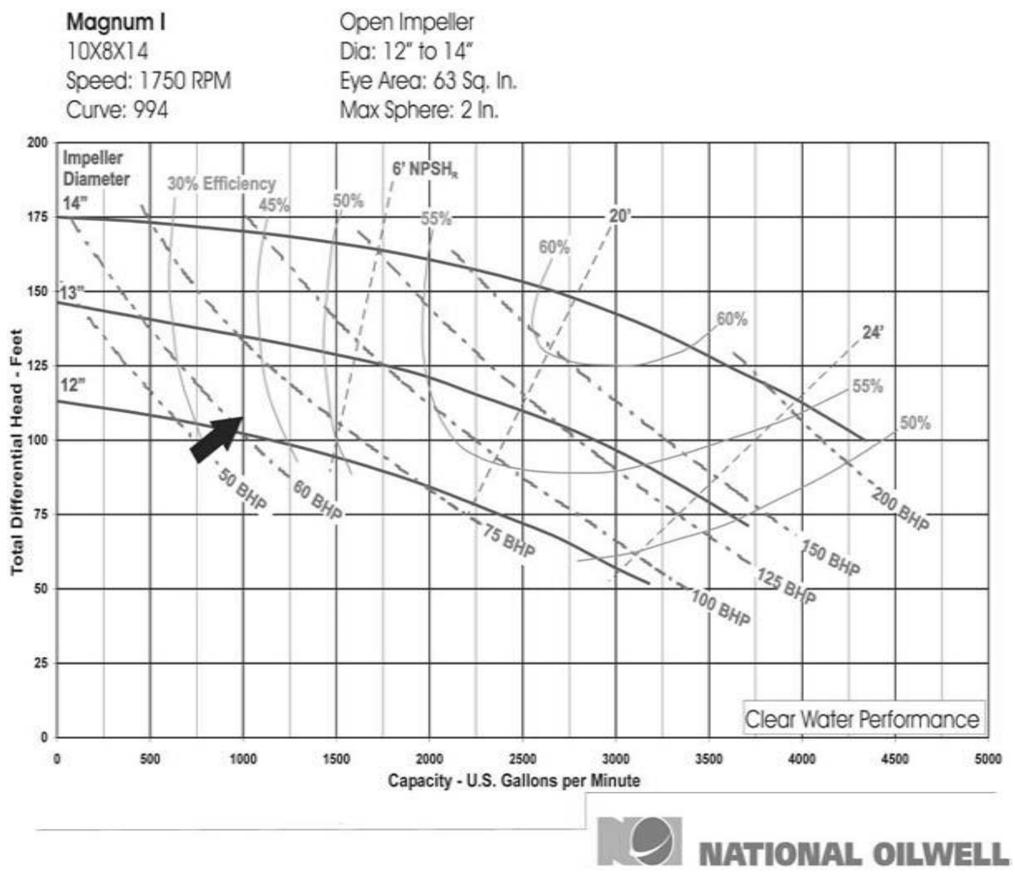


Figure: 10 x 8 x 14

With six different pumps that meet the criteria, it is important to select the pump that best meets the application. First consider where on the curve the operating point is located:

1. **Choice 5 x 4 x 14**: Located at the end of the curve requiring a very high NPSHR. If this pump is used, cavitation is likely due to insufficient NPSHA. But even given sufficient NPSHA exists and additional cones are added or the mud temperatures rises or if the desander cone wears, the pump could not handle the increased volume required. This would be an unacceptable sizing choice.

2. **Choice 6 x 5 x 11**: Most efficient pump for the application. An impeller diameter of 10.75 inches is near the maximum impeller diameter for this pump. If there were any miscalculation of friction losses or if the pump were to be relocated farther from the equipment, there would not be a way to significantly increase discharge head. This would be a fair choice.

3. **Choice 6 x 5 x 14**: Located in the center of the curve near the best efficiency point. The discharge head can be increased or decreased by changing the impeller size. NPSHR is low and the 6 x 5 x 14 sized pump costs less than a larger pumps. This would be an excellent choice.

4. **Choice 8 x 6 x 11**: Comments are the same as for 6 x 5 x 11.

5. **Choice 8 x 6 x 14**: Located left of the best efficiency point. This pump would perform well in the application but would require more energy than the 6 x 5 x14, resulting in higher operating costs. Some contractors may still wish to use this pump if they are utilizing a majority of 8 x 6 x 14's because they could reduce their spare-parts requirements. However, a different-size pump would only require an additional impeller and maybe casing held in stock. If space is available for spares, this is not a good reason to make this selection. This would be a good choice for this application.

6. **Choice 10 x 8 x 14:** Located at the very far left of the curve and a 10-inch suction line would have a line velocity below the recommended minimum one. Horsepower requirements are much higher than for other pumps. Although this pump would function in this application, it would be a bad choice.

The **6 x 5 x14** is the best pump for this application.
Knowing the SG of the fluid is necessary to determine the hp required. Maximum mud weight in this example is 16 lb/gal. To determine SG, use the following formula:

SG = $\frac{\frac{lb}{gal}}{8.34}$ therefore, $\frac{16}{8.34} = 1.92$ SG

where 8.34 lb = weight of 1 gallon of water.

Horsepower required:

SG x hp for water=1.92 x 39 = 74.88 hp:
A more accurate hp formula is as follows:
hp= $\frac{gpm\ x\ feet\ of\ head\ x\ SG}{3960\ x\ eff.}$
therefore
74.11 = $\frac{1000\ x\ 107\ x\ 1.92}{3960\ x\ .70}$

Motors are available in 75 hp, but it is advisable to select a 100-hp motor (if using an electric driver). A 100-hp motor would offer flexibility for the package and will compensate for any errors. It will increase flow rates against the pipe or equipment wear and will allow to exceed 16 lb/gal if future requirements dictate.

1.2.8 Net Positive Suction Head

NPSH is extremely important to the operation of a centrifugal pump. Factors that affect NPSH are atmospheric pressure, suction line friction loss, elevation, fluid temperature and SG. To calculate NPSHA:

$$NPSHA = \frac{(PAF - PVF)}{SG} \pm Z - SHF$$

where

PAF = atmospheric pressure, feet—at sea level is 34 feet of water. Therefore, if the supply tank is a vented or open-air tank at sea level, the atmosphere will apply 34 feet of pressure to the fluid surface. When the pump casing is filled with liquid and then activated, pressure at the eye of the impeller drops. Therefor the atmospheric pressure which is greater than this value will push the liquid into the pump suction. The pump does not suck fluid; it is the fluid that being pushed into the pump by the atmosphere.

PVF = vapor pressure, in feet—the amount of head required to maintain fluid in a liquid state. This value (for water) can be found in Table 1.2.8.a.

SG = specific gravity of fluid

SHF = suction head friction losses

Z = elevation or the liquid level above or below the pump centerline, in feet. If the supply tank is mounted on the same level as the pump, the elevation is the number of feet the fluid is above the centerline. Therefore, if the tank is 9 feet tall and when full has 8 feet 9 inches of liquid, then 8 feet can be added to the NPSHA equation as long as the tank is not going to be drained (centerline of the pump is 9 inches). If there is a desire to have the ability to drain the tank, this value should not be added to the equation. If the fluid level is below the pump centerline, this distance must be subtracted from the equation.

Properties of Water

Temperature		Vapor Pressure	
F	C	psi	ft
40	4.4	0.12	0.28
50	10	0.18	0.41
60	15.6	0.26	0.59
70	21.1	0.36	0.82
80	26.7	0.51	1.17
90	32.2	0.70	1.61
100	37.8	0.95	2.19
110	43.3	1.28	2.94
120	48.9	1.69	3.91
130	54.4	2.22	5.15
140	60	2.89	6.68
150	65.6	3.72	8.56
160	71.1	4.74	10.95
170	76.7	5.99	13.84
180	82.2	7.51	17.35
190	87.8	9.34	21.55
200	93.3	11.50	26.65
212	100	14.70	33.96

Table 1.2.8.a: Properties of water

To continue the previous example, the suction line will be 8 inches; flow rate 1000 gpm; fluid temperature 180˚ F; location is sea level; tank is open vented; liquid level above pump centerline is 8 feet and there is not a desire to drain the tank. The 8-inch-diameter suction line has one elbow, one branched tee and one butterfly valve and is 30 feet long:

Atmospheric pressure = 34 feet of water
Vapor pressure =? feet
SG = 1.92
Suction line friction losses previously calculated = 2.23 feet
Elevation = 8 feet

To determine vapor pressure of the fluid, refer to the properties of water in Table 1.2.8.a. Water-based drilling mud in 180˚F will require 17.9 feet of head to maintain fluid in a liquid state:

$$NPSHA = \frac{(PAF - PVF)}{SG} \pm Z - SHF$$

$$\text{NPSHA} = \frac{(34-17.35)}{1.92} + 8 - 2.23 = 14.44$$

Review the curve in Figure 1.2.8.b to determine NPSHR for the pump at the operating point (shown by the arrow). The NPSHR at the operating point is slightly less than halfway between 7 and 13 feet and would therefore equal 10 feet NPSHR. Because NPSHA is 14.44 and NPSHR is 10 feet, the pump will still have 4.44 feet of positive head that will prevent fluid from cavitating.
If NPSHA were less than or equal to NPSHR, the pump would cavitate and damage would occur.

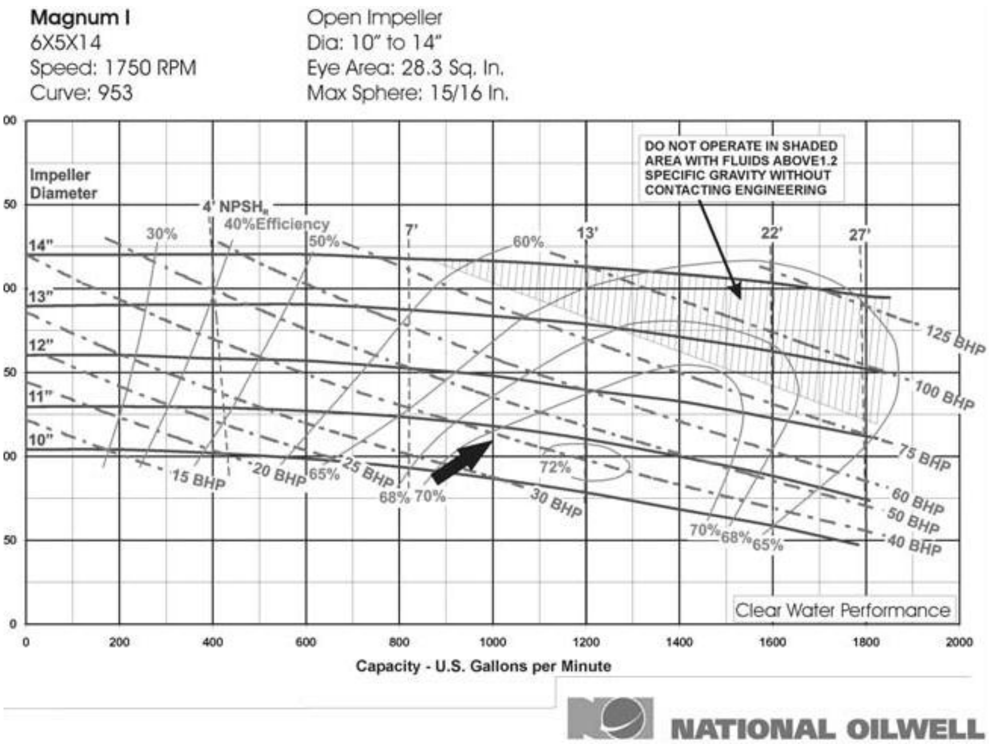


Figure 1.2.8.b: Centrifugal pump curve

Note that NPSH values are used only to determine whether adequate head will be maintained on the suction side of the pump to prevent cavitation. It does not however have any bearing on TDH required by the system.

Affinity Laws

If there is a known operating point and a different operating point is required, the following algebraic formulas can be used to accurately predict what changes should be made to alter flow or head and what the resulting horsepower requirements will be. A pump's performance can be altered by changing speed or by changing impeller diameter. Speed formulas are very reliable and impeller diameter formulas are accurate only for small variations in diameter.

Speed Formulas	or	Impeller Diameter Formulas (valid for small variations in dia. only, max 1″)
Flow:		
$\frac{gpm_1}{gpm_2} = \frac{rpm_1}{rpm_2}$	or	$\frac{gpm_1}{gpm_2} = \frac{Dia._1}{Dia._2}$
Total Differential Head:		
$\frac{TDH_1}{TDH_2} = \frac{(rpm_1)^2}{(rpm_2)^2}$	or	$\frac{TDH_1}{TDH_2} = \frac{(Dia_1)^2}{(Dia_2)^2}$
Horsepower:		
$\frac{hp_1}{hp_2} = \frac{(rpm_1)^3}{(rpm_2)^3}$	or	$\frac{hp_1}{hp_2} = \frac{(Dia_1)^3}{(Dia_2)^3}$

Figure 1.2.8.c: Affinity Laws

Friction Loss Formulas:

$$\frac{Friction\ loss1}{Friction\ loss2} = \frac{(gpm1)^2}{(gpm2)^2}$$

If a particular operating point and elevation of a system are known, it is possible to calculate a new operating point by using the following friction loss formulas. Assume that a system exists that has 20 feet of elevation and the pump is transferring water at 500 gpm and the pressure gauge reads 50 psi at the pump discharge. What head is required to produce 1000 gpm?

1. First convert psi to feet:

   Head = 50 psi x 2.31 / 1.0 SG
   Head = 115 feet

2. Subtract lift of 20 feet, since this is a constant:

115 feet head - 20 feet elevation = 95 feet of system friction loss at 500 gpm.

3. Utilize friction loss formulas to determine the new head required to produce 1000 gpm in this system:

$$\frac{Friction\ loss1}{Friction\ loss2} = \frac{(gpm1)^2}{(gpm2)^2} \quad \text{or} \quad \frac{X}{95} = \frac{(1000)^2}{(500)^2}$$

or X= 95 $\left(\frac{1000}{500}\right)^2$ = 380 feet

4. Add back lift: 380 + 20 = 400.

It would therefore be necessary to size a pump for 1000 gpm at 400 feet to obtain the desired flow rate of 1000 gpm in the existing system. It may be more economical to alter system discharge piping to reduce system friction losses than to pay power costs to produce 400 feet of head.

1.2.9 Recommended Suction Pipe Configurations

In addition to selecting the proper suction pipe diameter and having adequate NPSHA, the submergence level and suction pipe configuration must be considered.
Submergence level is the depth of the suction pipe inlet below the liquid surface. If an inadequate submergence level exists, an air vortex will form that extends from the liquid surface to the inlet of the suction pipe. This will introduce air into the system, resulting in either turbulent flow patterns or vapor locking of the pump.
The amount of submergence required varies with velocity of the fluid which is controlled by flow rate and pipe diameter. Refer to Figure 1.2.9.a. to determine submergence required based on fluid velocity.

(Fluid velocity can be found in Friction Loss Tables 1.2.6.a, in the column ‘‘V (ft/sec)’’).

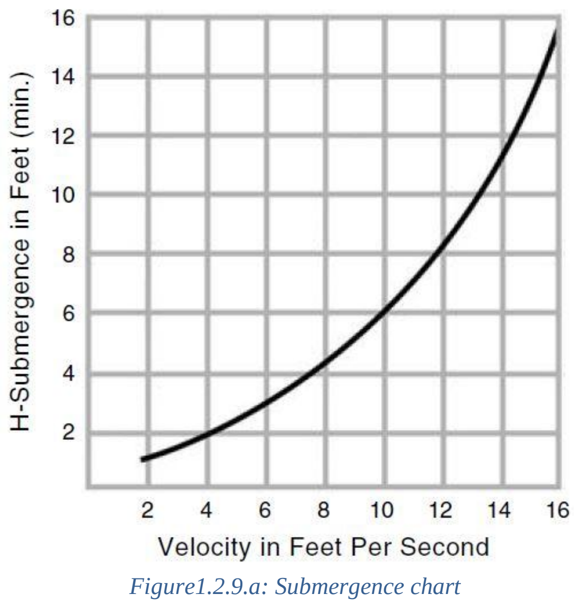


Figure1.2.9.a: Submergence chart

If a system utilizes a 6-inch suction line with a flow rate of 600 gpm, suction-line velocities will be 6.6 ft/sec and the line will therefore require approximately 3.5 feet of liquid surface above the suction-line entrance.
Once the submergence level drops below 3.5 feet, an air vortex will form, causing air to enter the pump suction, resulting in a turbulent flow pattern and/or vapor lock. A suction-line velocity of 6.6 ft/sec is ideal. Increasing the pipe diameter to 8 inches would result in an insufficient line velocity of 3.85 ft/sec. However, most systems will require the tank to have the ability to drain lower than 3.5 feet.One solution is to install a baffle plate over the suction pipe.

If a 14-inch baffle plate is installed the fluid velocities around the edge of the plate will only be 1.25 ft/sec which would allow the tank to be drained to approximately 1 foot above the suction pipe entrance. Refer to Figure 1.2.9.b for an illustrated view of a baffle plate.

In addition to proper line size and submergence level, a suction pipe should slope gradually upward from the source to the pump suction.
This prevents air traps within the suction line. There should be a straight run prior to the pump entrance of at least two pipe diameters in length to reduce turbulence. A smooth-flowing valve should be installed in the suction line that will allow the pump to be isolated for maintenance and inspection.
If a suction hose is used instead of hard piping, the hose must be non-collapsing. Refer to Figure 1.2.9.c.and 1.2.9.d for examples of accepted piping practices.

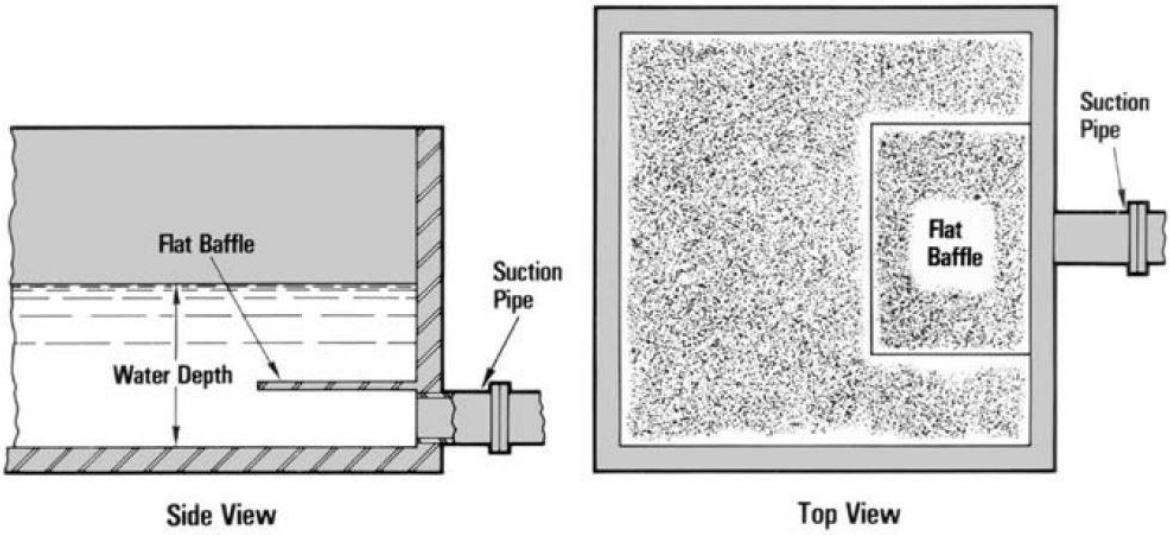


Figure 1.2.9.b.: Baffle plate illustration

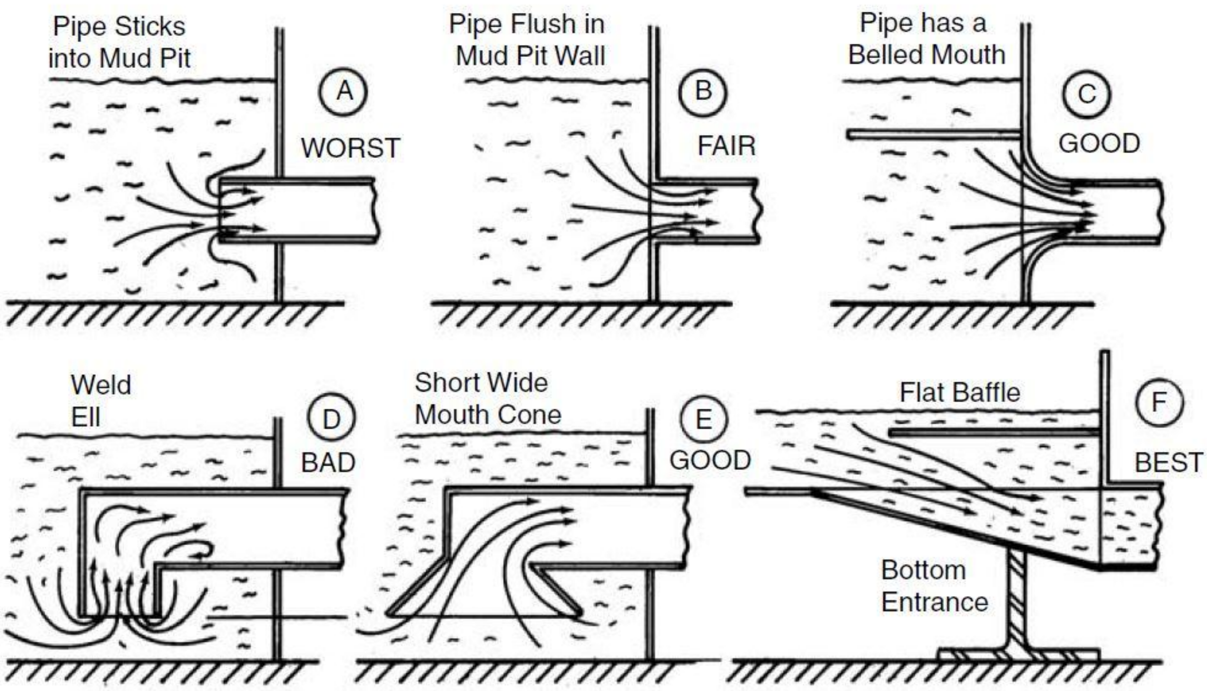


Figure 1.2.9.c: Suction line illustration

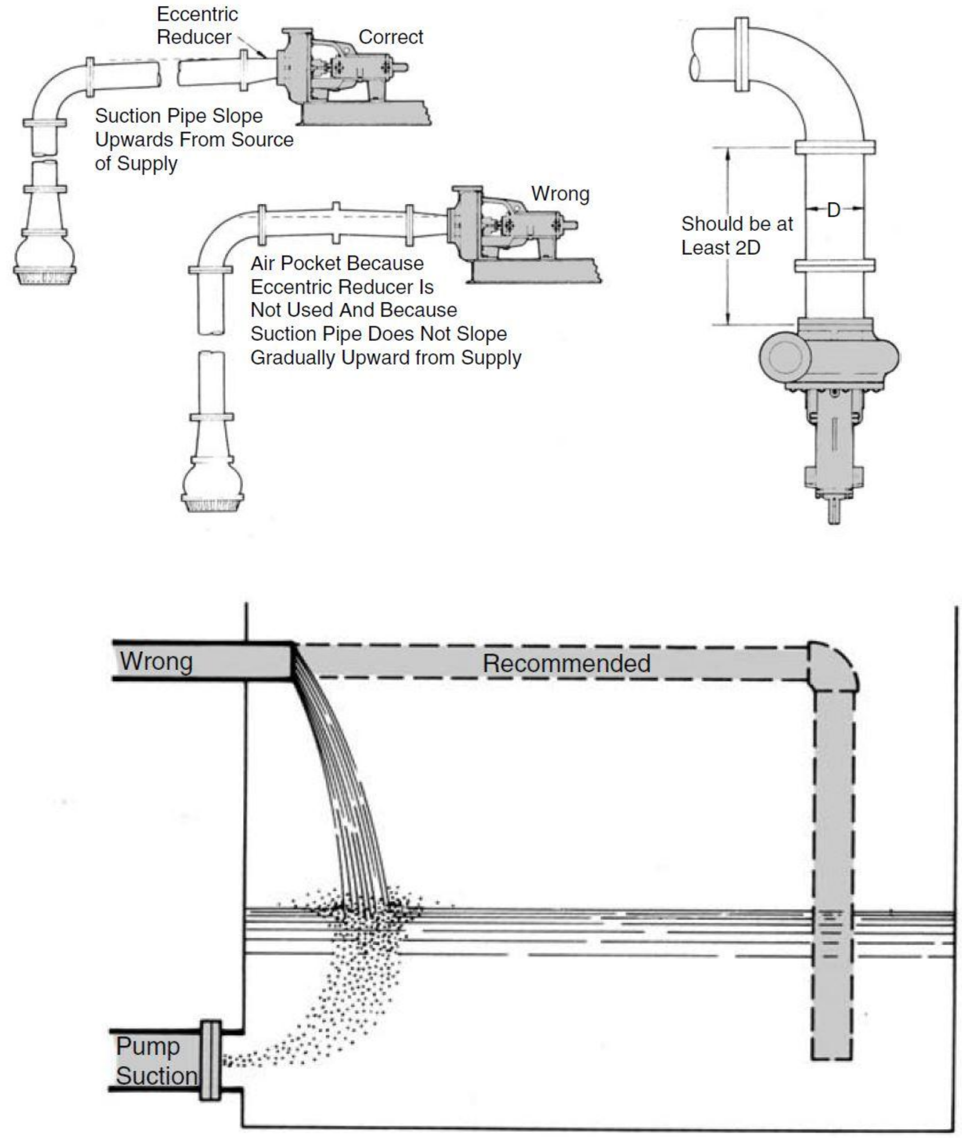


Figure 1.2.9.d: Piping recommendations

1.2.10 Centrifugal Set-up

Series Operation

There are times when a single centrifugal pump will not meet the head requirements of an application. Two pumps can be operated in series to achieve the desired discharge head, in which the discharge of one pump feeds the suction of the second pump.
The second pump boosts the head produced by the first. Therefore if an application required 2900 gpm at 200 feet of head, one option would be to run two 10 x 8 x 14 pumps in series. Each pump could be configured with a 13-inch impeller to produce 2900 gpm at 100 feet of head. When operated in series, the pumps would produce 2900 gpm at 200 feet of head.
This type of configuration is most commonly used for extremely long discharge runs. When running pumps in series, it is important not to exceed flange safety ratings. Additionally, it is not required to place pumps within close proximity of each other. If an application had a 6-mile discharge line the first pump could be located at the supply source and the second pump could be located 3 miles away.

Parallel Operation

Parallel operation is discouraged for centrifugal pumps. If an application exists that requires high volume and low head and volume required is greater than can be produced by a single pump, two pumps are sometimes used in a parallel configuration to meet the demand.
Two pumps that produce the same TDH can be configured so that each pump has an individual suction but both pumps feed into the same discharge line. If the pumps are identical, head in the discharge line is equal to that of the pumps but the volume is double what a single pump can produce.
However two centrifugal pumps will never have the exact same discharge head and as wear occurs one pump will produce less head than the other and the stronger pump will overpower the weaker pump and force fluid to backflow into the weaker pump.
For this reason parallel operation is not normally recommended.

Duplicity

Two pumps can be configured in parallel but only one pump is operated at a time thus providing a primary and a backup pump. The two pumps are separated by a valve in each discharge line that prevents one pump from pumping through the other. This type of configuration is perfectly acceptable and in crucial applications it is encouraged.

1.2.11 Standard rules for centrifugal pumps

I. Installation of a smaller suction line than the pump suction shouldn’t be allowed, as this can cause NPSH deprivation.
II. Pump suction valve mustn’t close or throttle while the pump is operating.
III. The pump should be primed and all air purge from the casing before starting, to prevent seal damage and vapor locking.
IV. Running a pump dry in unacceptable.
V. Pump discharge should always be above the pump suction, to prevent air from being trapped in the casing.
VI. NPSHA must be adequate.
VII. The insurance of adequate submergence of the suction line must be made, to prevent form air vortex.
VIII. Insurance of free of air fluid is being transferred.
IX. Suction and discharge piping should always lead upward. Lines that go up over and back down must be avoided, as this will cause air to become trapped in the line. If this is unavoidable, provide a means for bleeding air out of the line.
X. When sizing centrifugal pumps, work in feet is required, not psi.
XI. The pump driver must turn in the proper direction.
XII. Oil-lubricate bearings must be used for speeds above 2400 rpm.
XIII. Oil-lubricated pumps must be operated in a horizontal level position.
XIV. It is not allowed for the pump to support suction and discharge piping.
XV. Pump and driver must be properly aligned after all piping and positioning has been completed.
XVI. Proper sizing is desirable to produce head to meet the system requirements.
XVII. Sizing of the pump must be for the maximum flow rate, temperature and head that required.
XVIII. The suction line should be as short as possible.
XIX. The suction line should have a straight run at least two pipe diameters long directly in front of the pump suction.

Chapter 2 Drilling Pumps

2.0 Introduction

Drilling fluids are essential to the drilling process and they have various roles. They are needed to clean the rock fragments from beneath the bit and carry them to surface for farther treatment, testing and finally disposal. Moreover the hydrostatic column exerts sufficient hydrostatic pressure to prevent formation fluids flowing into the well and maintain stability of the borehole walls. Another important role is to cool and lubricate the drill string and bit.

The drilling fluid is stored in steel tanks located beside the rig and powerful pumps force the drilling fluid through surface high pressure connections to a set of valves called pump manifold, located at the derrick floor.

From the manifold, the fluid travels up the rig within a pipe called standpipe to approximately 1/3 of the height of the mast. From there the drilling fluid flows through a flexible high pressure hose to the top of the drill string. The flexible hose allows the fluid to flow continuously as the drill string moves up and down during normal drilling operations. The fluid enters in the drill string through a special piece of equipment called swivel located at the top of the kelly. The swivel permits rotation of the drill string while the fluid is pumped through the drill string.

The drilling fluid flows down the rotating drill string and jets out through nozzles in the drill bit at the bottom of the hole. It picks the rock cuttings generated by the drill bit action on the formation and subsequently flows up the borehole through the annular space between the rotating drill string and borehole wall.

There are different terms used to describe different sections of the well. The drill string volume can also be called “Surface to Bit”, this term describes the flow path that the mud will take when it is circulated. Generally you start at surface and pump mud to the bit. Annular volume can also be called “Bit to Surface” or “Bottoms Up”. Once mud reaches the bit it is then circulated back to surface.

Surface to Bit is shown as:

Bit to surface (Bottoms Up) is shown as:

There is another term used in the annulus and that is “Bit to Shoe”. The mud travels from the bit to the shoe, the Open-Hole section of the Annulus.

Bit to Shoe is shown as:

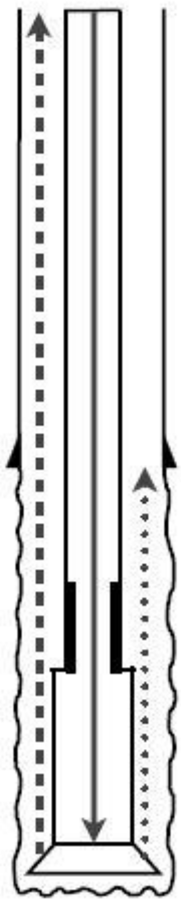


Figure2.a: Sections of the well

Process of mud circulation

The principal components of the mud circulation system are: pits or tanks, pumps, flow line, solids and contaminants removal equipment, treatment and mixing equipment, drill string, surface piping and valves.
The tanks are made of steel sheet and their size is usually 8-12 ft wide, 20-40 ft long and 6-12 ft high. The number of active tanks is determined by the size and depth of hole and usually 3 or 4; settling tank, mixing tank, suction tank contain a safe excess, neither too big nor too small of the total volume of the borehole. In the case of loss of circulation, this excess will provide the well with drilling fluid while the corrective measures are taken.
Above the tank level the drilling fluid flows through the flow line to a series of screens called the shale shaker. The shale shaker is designed to separate the cuttings from the drilling mud. The type of mud (i.e. oil-based or water-based) determines the type of the shaker required and the motion of the shaker. Deep holes require more than the customary three shakers.

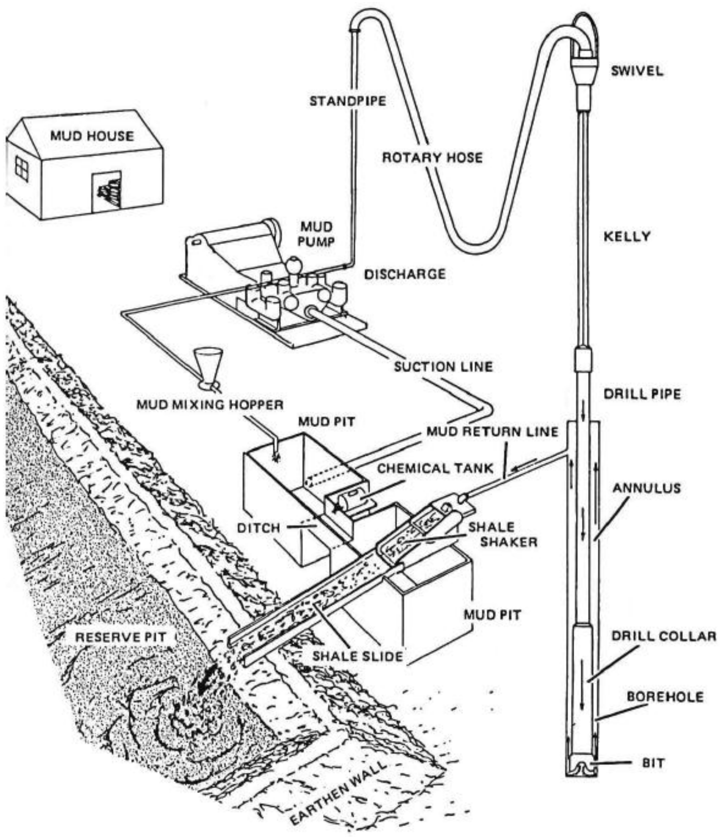


Figure2.b: Rig circulation system

Pump configurations

Rigs normally have two or three PDPs. During drilling of shallow portions of the hole, when the diameter is large, the two PDPs are connected in parallel to provide the highest flow rate necessary to clean the borehole. As the borehole deepens, less flow rate and higher pressure are required. In this case, normally only one PDP is used while the other is in standby or in preventive maintenance.

2.1 Positive Displacement Pumps (PDPs)

Types of the positive displacement pumps

The heart of the circulating system is the mud pumps. There are two types of PDP:

I. Duplex (2 cylinders)-double acting
II. Triplex (3 cylinders)-single acting

2.1.1 Duplex PDP

A double-acting duplex pump has two cylinders and pumps fluid on both the forward and backward stokes. As the piston moves forward discharging fluid ahead of it, the inlet port allows fluid to enter the chamber behind it. On the return the fluid behind the piston is discharged (i.e. on the rod side) while fluid on the other side is allowed in.

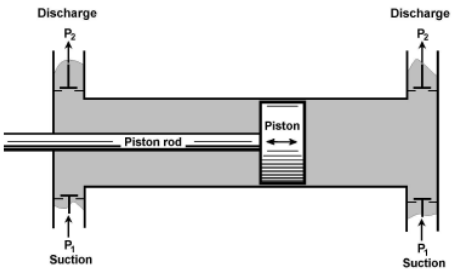


Figure2.1.1: Piston scheme (double action)

The theoretical displacement on the forward stroke is:

$$V_1 = \frac{\pi d^2 L}{4}$$

where,
d = liner diameter
L = stroke length

on the return stroke:

$$V_2 = \frac{\pi (d^2 - d_r^2) L}{4}$$

where,
d_r= rod diameter

Taking account of the 2 cylinders, and the volumetric efficiency Ev the total displacement (in gallons) of one pump revolution is:

$$2(V_1 + V_2)E_v = \frac{2\pi(2d^2 - d_r^2)LE_v}{4}$$

The pump output can be obtained by multiplying this by the pump speed in revolutions per minute. (In oilfield terms 1 complete pump revolution = 1 stroke, therefore pump speed is usually given in strokes per minute) e.g. a duplex pump operating at a speed of 20 spm means 80 cylinder volumes per minute. Pump output is given by:

$$Q = \frac{(2d^2 - d_r^2)LE_v R}{147}$$

where,
Q = flow rate (gpm)
d = liner diameter (in.)
dr = rod diameter (in.)
L = stroke length (in.)
R = pump speed (spm)

2.1.2 Triplex PDP

A single-acting triplex pump has three cylinders with suction during one direction of piston motion and discharge on the other.

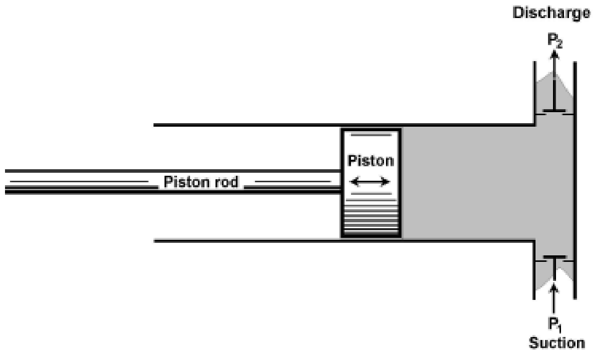


Figure2.1.2.a: Piston scheme (single action)

The discharge volume for one pump revolution is:

$$3V_1E_v = \frac{3\pi d^2 L E_v}{4}$$

The pump output is found by multiplying by the pump speed:

$$Q = \frac{d^2 L E_v R}{98.03}$$

where,
Q = flow rate (gpm)
L = stroke length (in.)
d = liner diameter (in.)
R = pump speed (spm)

More power can be delivered using a triplex pump since higher pump speeds can be used. They will also produce a smoother discharge since they pump an equal volume at every 120 degree rotation of the crankshaft.

Triplex pumps are generally used in offshore rigs and duplex pumps on land rigs. Triplex pumps have the advantages of being lighter, give smoother discharge and have lower maintenance costs.

Pump liners

Pump liners fit inside the pump cavity. These affect the pressure rating and flow rate from the pump. For a given pump, a liner has the same OD but with different internal; diameters. The smaller liner (small ID) is used in the deeper part of the well where low flow rate is required but at much higher operating pressure. The size of the pump is determined by the length of its stroke and the size of the liner.

Surge Dampeners

Due to the reciprocating action of the PDPs, the output flow rate of the pump presents a "pulsation" (caused by the changing speed of the pistons as they move along the liners). This pulsation is detrimental to the surface and down hole equipment (particularly with MWD pulse telemetry system). To decrease the pulsation, surge dampeners are used at the output of each pump.
A flexible diaphragm creates a chamber filled with nitrogen at high pressure. The fluctuation of pressure is compensated by a change in the volume of the chamber. A relief valve located

in the pump discharge line prevents line rupture in case the pump is started against a closed valve.

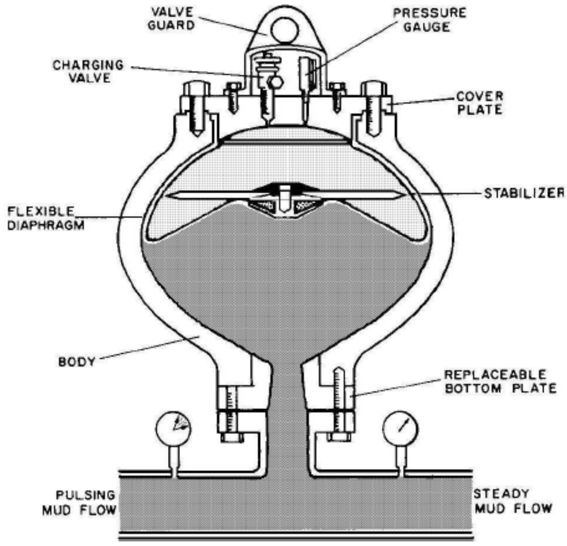


Figure2.1.2.b: typical surge dampener

2.1.3 Volume Calculations for Triplex PDPs

The most common type of rig pump used in drilling is the triplex pump. It works by using a tight-fitting piston to force the mud through a liner:

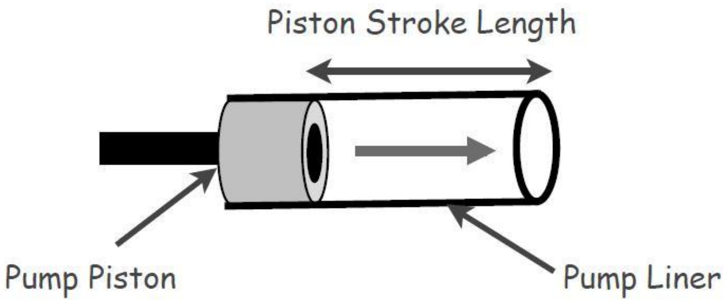


Figure2.1.3.a: Single piston

The pump liner is a cylinder and we must calculate the capacity of this cylinder:

Capacity (bbl/ft) $=\frac{d^2\text{ (in)}}{1029.4}$

The barrels per foot output of the cylinder where d is the ID of the pump liner, will work if the piston stroke length is one foot (12").

Example:

The liner output for a pump with a stroke length of 12" and an ID of 6" is:

Pump Liner Output = 36 ÷ 1029.4 = 0.035 bbl

In case of different pump stroke we use the formula:

Pump Liner Output (bbl) = d^2 (in) ÷ 1029.4 x stroke length (ft)

The liner output for a pump with a stroke length of 10" and an ID of 6" is:

Pump Liner Output = 36 ÷ 1029.4 x 0.833 = 0.029 bbl

A Triplex Pump has three liners and pistons all working at the same time.

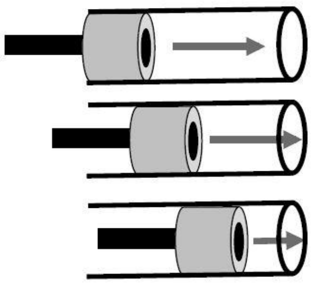


Figure2.1.3.b: Triple piston

In field units pump output is expressed as barrels per stroke (bbl/stk). One stroke is completed when all three liners have emptied. Writing this as a formula we get:
Pump Output (bbl/stk) = d^2 (in) ÷ 1029.4 x stroke length (ft) x 3
Example:
The liner output for a pump with a stroke length of 12" and an ID of 6" is:
Pump Output = 36 ÷ 1029.4 x 1 x 3 = 0.105 bbl/stk

Pump operating pressure

The horse power requirements of the pump depend on the flow rate and the pressure. The operating pressure depends on flow rate, depth and size of hole, size of drill pipe and drill collars, mud properties and size of nozzles used. A full hydraulics program needs to be calculated to determine the pressure requirement of the pump.

Pump Power

Pumps convert mechanical power into hydraulic power. From the definition of power: $P = F * u$
In its motion, the piston exerts a force F on the fluid that is equal to the pressure differential in the piston Δ_p times the area A of the piston, and the velocity u is equal to the flow rate q divided by the area A, that is:

$$PH = (\Delta p * A) * \frac{q}{A} = \Delta p * q$$

For PH in hp, p in psi, and q in gal/min we have:

$$PH = \frac{\Delta p * q}{1714.29}$$

Pumps generally do not work at 100% efficiency and as a result you will not get 100% of the theoretical volume discharged.

Parameters affecting the flow rate

The great flexibility in the pressure and flow rate is obtained with the possibility of changing the diameters of the pair piston–liner. The flow rate depends on the following parameters:

- Stroke length L_s (normally fixed)
- Liner diameter d_L
- Rod diameter d_R (for duplex PDP only)
- Pump speed N (normally given in strokes/minute)
- Volumetric efficiency E_v of the pump

2.1.4 Quintuplex PDP

A single-acting quintuplex pump has five cylinders with suction during one direction of piston motion and discharge on the other.

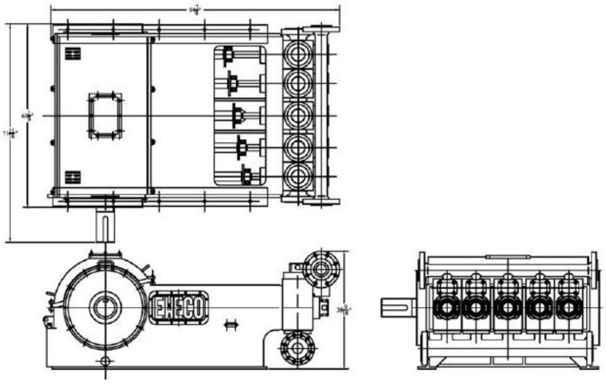


Figure2.1.4.: EQ-750 Quintuplex pump, Ellis Williams Engineering Compan

2.1.5 Quintuplex Vs Triplex PDP

E series pump design

The E-2200 mud pump is a triplex single acting horizontal design (conventional) mud pump with the individual fluid cylinders and rod assembly centerlines oriented in horizontal planes. This type of pump is the most accepted type of reciprocating pump found on land and off shore drilling rings in the industry today.

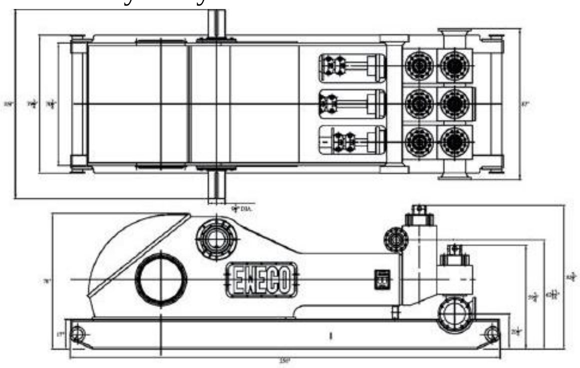


Figure 2.1.5a: E-2200

E-2200 Performance Ratings								
E-2200		Strokes/min (spm)	105	100	95	90	80	75
		Input Power (HP/*kW*)*	2214 *1652*	2109 *1573*	2003 *1495*	1898 *1416*	1687 *1259*	1582 *1180*
Piston Size (in.)	Rated Pressure (psi/*Mpa*)	Displacement (gal/rev)	Output (gal/min / *barrels per minute*)					
9	2625 *18.1*	12.393	1301 *31.0*	1239 *29.5*	1177 *28.0*	1115 *26.6*	991 *23.6*	929 *22.1*
8	3325 *22.9*	9.792	1028 *24.5*	979 *23.3*	930 *22.1*	881 *21.0*	783 *18.7*	734 *17.5*
7	4325 *29.8*	7.497	787 *18.7*	750 *17.8*	712 *17.0*	675 *16.1*	600 *14.3*	562 *13.4*
6	5900 *40.7*	5.508	578 *13.8*	551 *13.1*	523 *12.5*	496 *11.8*	441 *10.5*	413 *9.8*
5.5	7025 *48.4*	4.628	486 *11.6*	463 *11.0*	440 *10.5*	417 *9.9*	370 *8.8*	347 *8.3*
5	7500 *51.7*	3.825	402 *9.6*	382 *9.1*	363 *8.7*	344 *8.2*	306 *7.3*	287 *6.8*

Figure 2.1.5.b: E-2200 (three cylinder pump)

EQ series pump design

The EQ-2200 mud pump is a quintuplex single acting horizontal design (conventional) mud pump with the individual fluid cylinders and rod assembly centerlines oriented in horizontal planes.

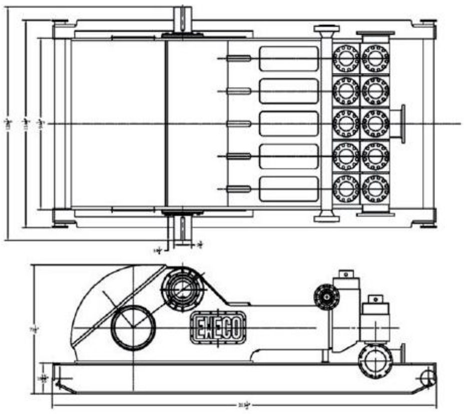


Figure 2.1.5.c: EQ-2200

EQ-2200 Performance Ratings								
EQ-2200		Strokes/min (spm)	135	130	120	110	100	90
		Input Power (HP/*kW*)*	2239 *1671*	2156 *1609*	1991 *1485*	1825 *1361*	1659 *1237*	1493 *1114*
Piston Size (in.)	Rated Pressure (psi/*Mpa*)	Displacement (gal/rev)	Output (gal/min / *barrels per day*)					
7.5	2230 *15.4*	11.475	1549 *36.9*	1492 *35.5*	1377 *32.8*	1262 *30.1*	1147 *27.3*	1033 *24.6*
6.5	2975 *20.5*	8.619	1164 *27.7*	1120 *26.7*	1034 *24.6*	948 *22.6*	862 *20.5*	776 *18.5*
6	3475 *24.0*	7.344	991 *23.6*	955 *22.7*	881 *21.0*	808 *19.2*	734 *17.5*	661 *15.7*
5.5	4125 *28.4*	6.171	833 *19.8*	802 *19.1*	741 *17.6*	679 *16.2*	617 *14.7*	555 *13.2*
5.25	4550 *31.4*	5.623	759 *18.1*	731 *17.4*	675 *16.1*	619 *14.7*	562 *13.4*	506 *12.0*
5	5000 *34.5*	5.100	688 *16.4*	663 *15.8*	612 *14.6*	561 *13.4*	510 *12.1*	459 *10.9*

Figure2.1.5.d: EQ-2200 (five cylinder pump)

Specification Comparison

Specification	**EQ-2200**	**E-2200**
Stroke length in. (cm)	12’’ (30.48)	15’’ (38.10)
Maximum Rated Pump Speed-SPM	135	105
Bare Pump Weight - lbm(kgs)	67,500 (30,617)	85,000 (38,555)
Footprint - sq ft. (sm)	189 (17.5)	191.28 (17.7)
Maximum Working Pressure –PSI (Mpa)	7500 (51.71)	7500 (51.71)
Maximum Flow Rate-GPM (lpm)	1549	1301
Number of main bearings	4	2

Figure2.1.5.e: Specification Comparison

Piston Velocity

As reflected in the above table, the stroke of the E series pump exceeds that of the EQ series pump stroke by 3 inches. However, overall piston speed values of the two series are similar to one other. Piston speed for the EQ is 135 FPM and 131.25 FPM for the E pump. Piston speed can be calculated as:
Stroke length (in) x Maximum rate SPM / 12 (in) = Piston speed in FPM

Weight

Bare pump weight does not include any ancillary equipment such as pulsation dampeners, discharge crosses, pressure relief valves, mud gauges, centrifugal charge pumps or master skids. The reduced weight of the EQ vs. the E and other similar sized pumps will not require contractors to increase structural support when upgrading existing rigs from triplex pumps to quintuplex pumps.

Footprint

The EQ's small footprint allows it to be used on existing offshore rigs without having to construct additional space on board.

Measurement While Drilling /Logging While Drilling operations

During MWD/LWD operations the use of the EQ series provides additional value when compared to the E series. Pulsation signatures of five cylinders pumps occur at higher frequencies than triplex pumps. The higher frequency pulsation signature allows for wider frequency bands to transmit MWD/LWD data up to the well bore annulus, thereby increasing data transfer rates during MWD/LWD.

To calculate the dominant frequency for a quintuplex pump operating at 60 SPM:

60 (SPM) x 10 (pulse peaks) per pump stroke / 60 (seconds) = pulse signature (10 Hz)

To calculate the dominant frequency for a triplex pump operating at 60 SPM:

60 (SPM) x 6 (pulse peaks) per pump stroke / 60 (seconds) = pulse signature (6 Hz)

The dominant frequency of the EQ operating at 60 strokes per minute produces a 10 Hz frequency pulse signature and the dominant frequency of a triplex pump operating at 60 strokes per minute produces a 6 Hz signature. The frequency of the quintuplex signature is increased 40 % over the triplex operating at the same speed.

The higher frequency of the quintuplex produces a signature that allows MWD/LWD companies to raise their frequency transmission which translates into higher data rate increase.

Utilizing current proven conventional triplex pumps with the addition of two more cylinders provides an increased performance envelope, reduces pulsations on the rig structure and piping, and substantially increase the data-rate transfer bandwidth during MWD/LWD (measurement while drilling/logging while drilling) operations.

Current data rate transfer for MWD/LWD when using reciprocating pumps are limited to frequencies below the first dominant frequency. Using a comparable horsepower quintuplex pump increases the data transfer rate 66% over current triplex pumps. Due to the shorter stroke of a comparable quintuplex pump, the pump can also be operated at higher stroke rates that produce an additional 33% gain in frequency bandwidth. This is achieved through increased frequency of the additional pump cylinders and the ability to operate the pump at higher pump speeds.

The inherent design attributes of conventional quintuplex pumps reduce fluid flow variations (pulse) versus comparable horsepower triplex pumps. The five operating cylinders are timed

at 72 ° whereas conventional triplex pumps are timed at 120 ° intervals per 360 ° of pump rotation. The additional cylinders reduce flow variations of quintuplex to 7% versus a 23% flow variation for conventional triplex pumps.
Thought both series have equivalent horsepower, in contrast, the pulse signature of the EQ series is greatly reduces compare to that of the E series, thereby increasing the performance envelope for MWD/LWD operations. In addition weights and dimensional envelopes of the EQ series are less than the E series.

2.2 Centrifugal Pumps & Standard Drilling Equipment

Hoppers, mud guns, desanders, desilters, degassers and more drilling equipment as and triplex pumps requiring supercharging and all have one thing in common:
They require 76–80 feet of inlet head to operate as designed. Exceptions do exist and the equipment manufacturer should be consulted.
Since most applications in drilling systems require 80 feet of head at the inlet of the equipment, this simplifies the job of sizing centrifugal pumps.
Following are standard flow rates when equipment has an 80-foot inlet head:

- 6-inch hopper with standard 2-inch nozzle: 550 gpm
- 4-inch hopper with 1 1/2-inch nozzle: 300 gpm
- Mud gun with 3/4-inch nozzle: 85 gpm per gun
- Mud gun with 1-inch nozzle: 150 gpm per gun
- Desander/desilter 4-inch cone: 60 gpm per cone
- Desander/desilter 10-inch cone: 500 gpm per cone
- Degasser: 600 gpm

These are general standards and all values, both suction and discharge conditions should be verified and consider prior to sizing the centrifugal pump.

Supercharging Mud Pumps

Triplex mud pumps are often operated at speeds at which head in the suction tank is insufficient to maintain fluid against the piston face during the filling stroke. If fluid does not remain against the face, air is sucked in from behind the piston, causing a fluid void. If a void is formed, the piston strikes the fluid when the piston reverses direction during the pressure stroke. This causes a shock load that damages the triplex power end and fluid end and lowers expendable parts life.
Supercharging pumps are used to accelerate fluid in the suction line of a triplex mud pump during the filling stroke, allowing fluid to maintain pace with the piston.
A properly sized supercharging pump will accelerate fluid so that fluid voids and shock loads do not occur.
Triplex mud pumps normally have shock loads at speeds greater than 60 strokes per minute (spm) (when not supercharged). Without proper equipment this would go unnoticed until the pump exceeded 80 strokes per minute and meanwhile the shock load is damaging the pump.
Supercharging requires an oversized pump with wide impellers to adequately react to rapid changes in flow required by the triplex mud pump.
When sizing a centrifugal pump for a mud pump supercharging application the pump should be sized for 1 1/2 times the required flow rate. Therefore if the triplex mud pump maximum flow rate is 600 gpm, the centrifugal pump should be sized for 900 gpm.
High-speed piston and plunger pumps that stroke above 200 spm should be designed with a supercharging pump that produces 1 3/4 to 2 times the required flow rate.

Supercharging is one of the few applications in which the centrifugal pump does not have steady flow. Small impellers operating at 1750 rpm have a tendency to slip through the fluid when acceleration is needed. This is similar to car tires slipping on wet pavement. Even though it sometimes appears that the small impeller running at 1750 rpm is providing enough head, shock loading may be occurring. Supercharging pumps should have larger impellers running at either 1150 (60 cycles) or 1450 rpm (50 cycles) and should normally be sized to produce 85 feet of head at the triplex suction inlet.

Supercharging pumps should be located as close to the supply tank as possible. Mounting supercharging pumps near the triplex and away from the supply tank will transfer suction problems from the triplex to the centrifugal pump. If the centrifugal pump does not have a favorable supply with short suction run, it will have an insufficient supply to accelerate fluid.
Piping for supercharging pumps and triplex pump suctions should be oversized for the flow rate. Pipe should be sized so the change in line velocity during pulsations will not be over 1.5 ft/sec during the change from low flow rate to high flow rate during the triplex pulsation cycles.

Proper Size of Centrifugal Pumps

Many factors affect the performance of a centrifugal pump and must be considered during pump selection. During rig design, the centrifugal pump is often considered a low-cost product that does not warrant a great deal of engineering consideration.
Many times the centrifugal is sized and ordered based on existing packages utilized on other rigs. This can cause serious problems because each rig has unique operating conditions and piping designs.
Centrifugal pumps are used for a variety of applications and feed other, much more expensive equipment. If the centrifugal pumps are not properly sized, they and other equipment can be adversely affected. Proper sizing, design and installation of centrifugal pumps can directly affect the efficiency and operating cost of the rig.

2.3 Friction Losses Calculations

Hydraulic friction in the circulating system

Head loss Δp_f

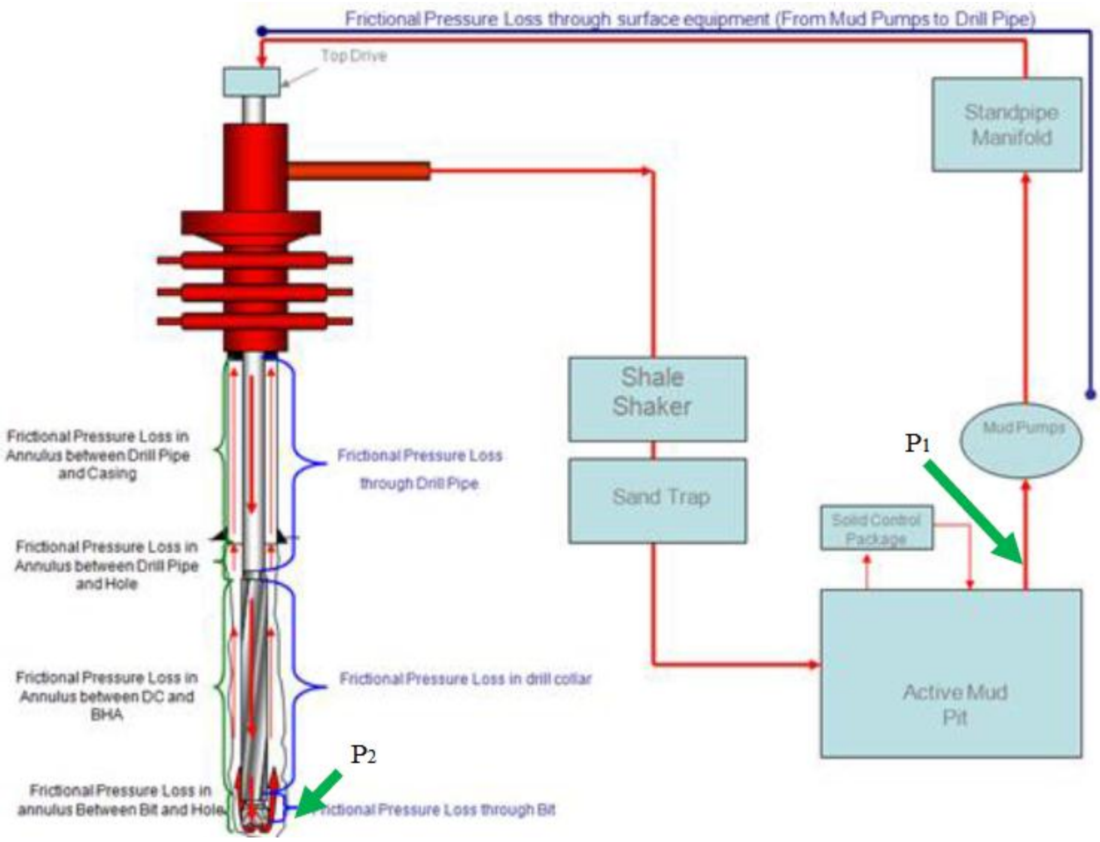
$$\frac{p_1}{\rho g}+\frac{v_1^2}{2g}+z_1+\Delta p_p=\frac{p_2}{\rho g}+\frac{v_2^2}{2g}+z_2+\Delta p_f$$

From left to right; pressure head, velocity head, gravity or potential energy head and pump head. To the far right the friction loss head is added and referred to as head-loss

After customized the energy balance equation for drill mud fluid flow and simplifying in field units:

$$p_2 = p_1 + 0.052\rho(D_2 - D_1) - 8.074 * 10^{-4}\rho(u_2^2 - u_1^2) + \Delta p_p - \Delta p_f$$

A significant amount of power is required to overcome the frictional resistance to flow of the fluid in the drillstring, annulus and through the nozzles in the bit. The *hydraulic power* which is expended when circulating the fluid is a direct function of the pressure losses and the flowrate through the system. Since the flowrate through all parts of the system is equal, attention is generally focused on the pressure losses in each part of the system. The pressure required to circulate the fluid through the drillstring and annulus are often called *sacrificial pressure losses*, since they do not contribute anything to the drilling process but cannot be avoided if the fluid is to be circulated around the system.



P_1= pump inlet pressure
P_2= bottom hole pressure
D_1= initial depth
D_2= final depth
u_1= initial velocity
u_2= final velocity
Δp_p= pressure added due to pump
Δp_f = pressure loss due to friction
Δp_f is the summation of the following friction losses:

- Friction pressure losses through surface equipment (from mud pumps to drill pipe)
- Friction pressure losses through drill pipe
- Friction pressure losses in drill collars
- Friction pressure losses through drill bit
- Friction pressure losses in annulus between bit and hole
- Friction pressure losses in annulus between drilling collars and bottom hole assembly
- Friction pressure losses in annulus between drill pipe and hole
- Friction pressure losses in annulus between drill pipe and casing

The principal factors which influence the magnitude of the pressure losses in the system are:

- The geometry of circulating system (e.g. I.D. of drillpipe, length of drillpipe)
- The flowrate through the system
- The flow regime in which the fluid is flowing (laminar/turbulent)
- The rheological properties of the circulating fluid

Calculations of friction losses in pipes and annuli, for laminar and turbulent flow, for three different rheology models: Newtonian, Bingham Plastic and Power-law.

Laminar flow

Converting to field units we have the equation for the pressure loss for the flow of a Newtonian Fluid in a pipe:

$$\frac{dP_f}{dL} = \frac{\mu\overline{v}}{1500d^2}$$

in an annulus:

$$\frac{dP_f}{dL} = \frac{\mu\overline{v}}{1000(d_2 - d_1)^2}$$

The equation for the frictional pressure loss for the circulating of a Bingham Plastic Fluid in a pipe:

$$\frac{dP_f}{dL} = \frac{\mu_p\overline{v}}{1500d^2} + \frac{\tau_y}{225d}$$

in an annulus:

$$\frac{dP_f}{dL} = \frac{\mu_p\overline{v}}{1000(d_2 - d_1)^2} + \frac{\tau_y}{200(d_2 - d_1)}$$

The equation for the frictional pressure loss for the circulating of a Power law Fluid in a pipe:

$$\frac{dP_f}{dL} = \frac{K\bar{v}}{144{,}000d^{(1+n)}}\left(\frac{3 + {}^{1}\!/_{n}}{0.0416}\right)^n$$

in an annulus:

$$\frac{dP_f}{dL} = \frac{K\bar{v}}{144{,}000(d_2 - d_1)^{(1+n)}}\left(\frac{2 + {}^{1}\!/_{n}}{0.0208}\right)^n$$

Turbulent flow

The equation for the pressure losses of a Newtonian fluid in a pipe is:

$$\frac{dP_f}{dL} = \frac{f\rho\bar{v}^2}{25.8d}$$

in an annulus:

$$\frac{dP_f}{dL} = \frac{f\rho\bar{v}^2}{21.1(d_2 - d_1)}$$

The equation for the pressure losses of a Bingham Plastic Fluid in a pipe is:

$$\frac{dP_f}{dL} = \frac{f\rho\bar{v}^2}{25.8d}$$

in an annulus:

$$\frac{dP_f}{dL} = \frac{f\rho\bar{v}^2}{21.1(d_2 - d_1)}$$

The equation for the pressure losses of a Power law Fluid in a pipe is:

$$\frac{dP_f}{dL} = \frac{f\rho\bar{v}^2}{25.8d}$$

in an annulus:

$$\frac{dP_f}{dL} = \frac{f\rho\bar{v}^2}{21.1(d_2 - d_1)}$$

Chapter 3 Production Pumps

3.0 Introduction

Hydrocarbons flow to the surface under natural flow when the discovery well is completed in a virgin reservoir. The fluid production resulting from reservoir development will normally lead to a reduction in the reservoir pressure, increase in the fraction of water being produced together with a corresponding decrease in the produced gas fraction.
All the above factors will reduce or may even stop the flow of fluids from the well. The goal is to install within the well completion some form of artificial lift, gas lift or pump lift, which will add energy to the well fluid and will allow the well to flow at an economic production rate.

It has been estimated that in 1994 more than 900,000 producing wells existed worldwide. Only 7% of these flowed naturally while the remaining 93% required some form of artificial lift to increase their average production, which at that time were less than 70 bpd.

Down hole pumps are being used to boost the productivity of a well by lowering the bottom hole flowing pressure. They do that by increasing the pressure at the bottom of the tubing with a sufficient amount to lift the liquid stream to the surface and in this way the bottom hole pressure will eventually decreased. The pressure gradient is higher in a pumped well than it would be without the pump, because the most of the gas produced with the liquid is vented through the casing-tubing annulus.
The process in which bottom hole pressure is decreased in gas lift follows the opposite way of that of the pump lift. Gas lift lowers the pressure gradient in the tubing and finally the bottom hole pressure is reduced.

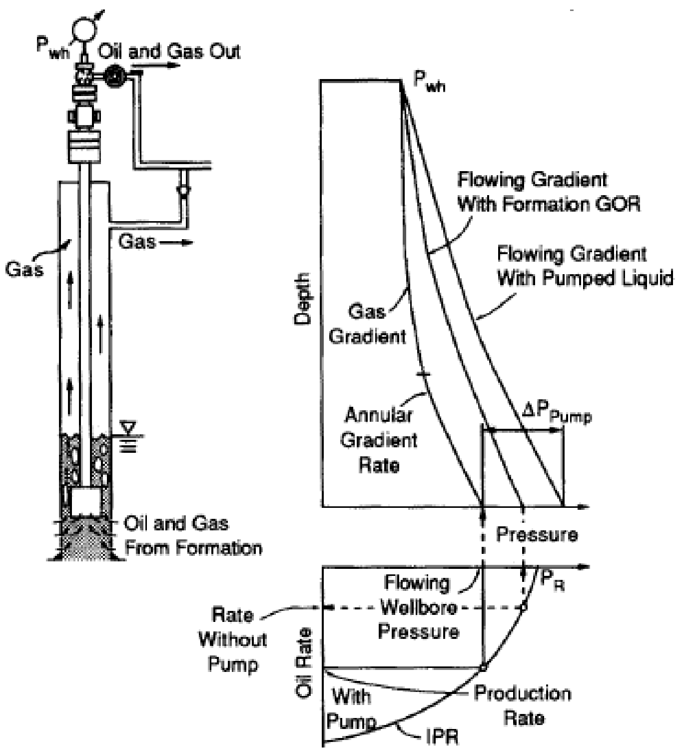
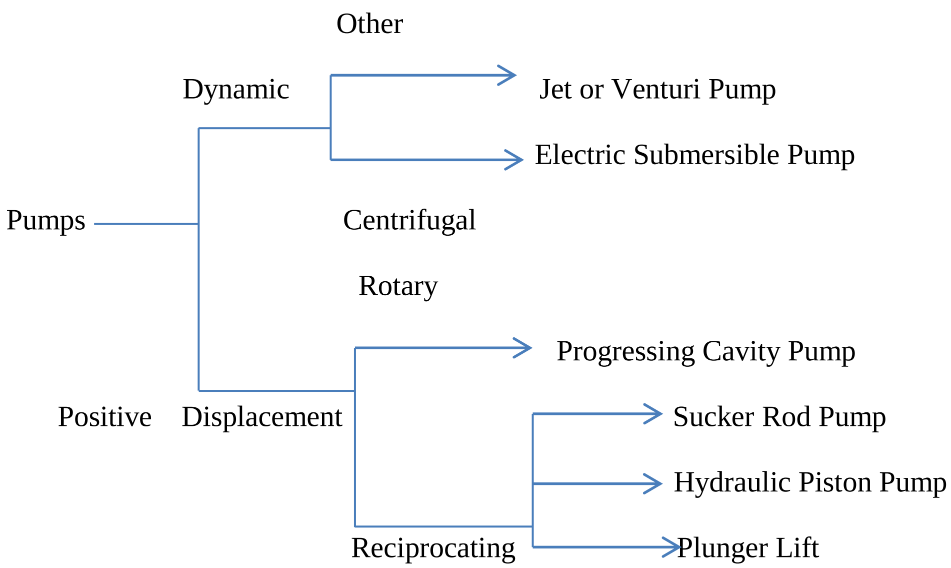


Figure: Well configuration and pressure profiles for an oil well (After Golan and Whitson, 1991)

Types of pumps in reservoir production

Both types of pumps are being used also in this aspect of the industry: positive displacement pumps and dynamic displacement pumps.



The need for artificial lift

Artificial lift must be applied when a well is no longer flowing or the production rate is not economic. Figure 3.a illustrates such a situation where the reservoir pressure is so low that the static fluid level is below the wellhead and no flow to the surface occurs.

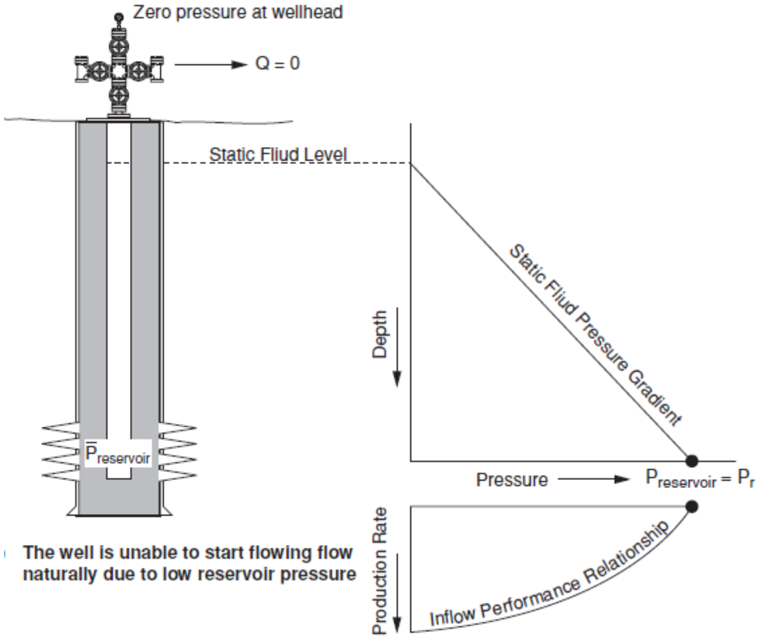


Figure 3.a: The well is unable to initiate natural flow

Pump Installation

The installation of a pump near the static fluid level allows a small drawdown Δp_1 to be created which will help the well to start flow at rate q_1. The static and flowing pressure gradients in figures 3.a & 3.b are similar since frictional pressure losses in the tubing are small at this low q_1 flow rate. The same production rate will occur when the pump is relocated to the bottom of the tubing, provided the pressure drop across the pump and hence the drawdown remains the same. The advantage of placing the pump near the perforations is that the maximum potential production can now be achieved (Figure 3.c) by imposing a large drawdown Δp_2 on the formation and "pumping the well-off" by producing the well at q_2, a flow rate that is slightly smaller than the AOF (Absolute Open Flow).
Artificial lift design requires the pump that will be installed is matched to the well inflow (IPR curve) and outflow performance (VLP curve).

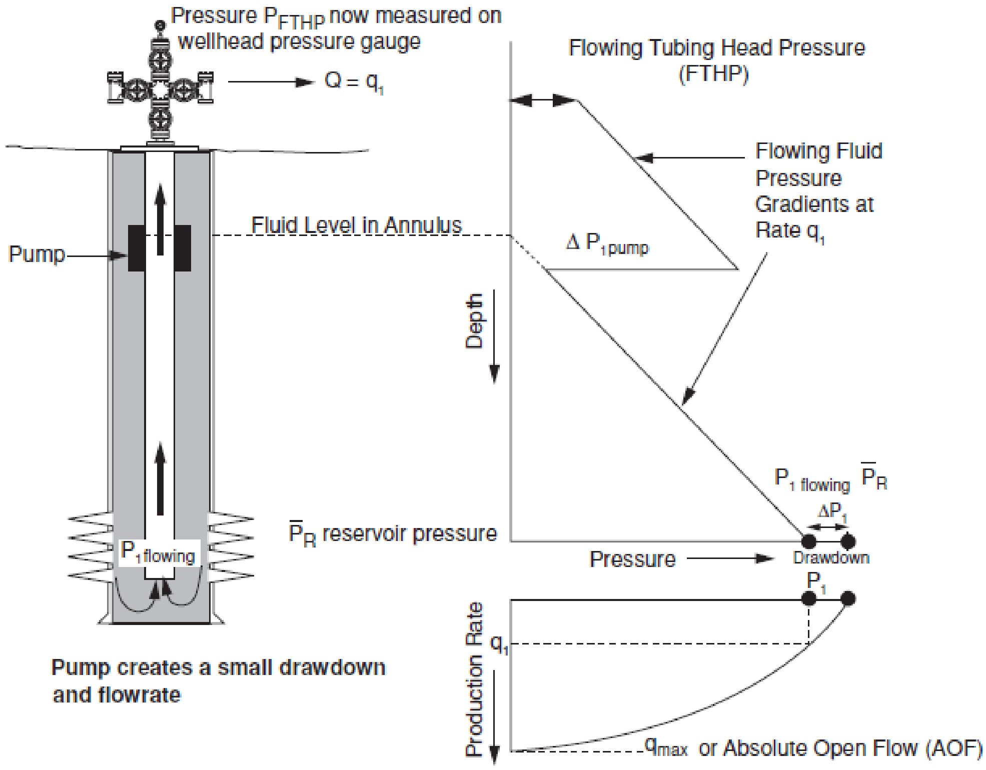


Figure 3.b: Pump creates a small drawdown and flow rate

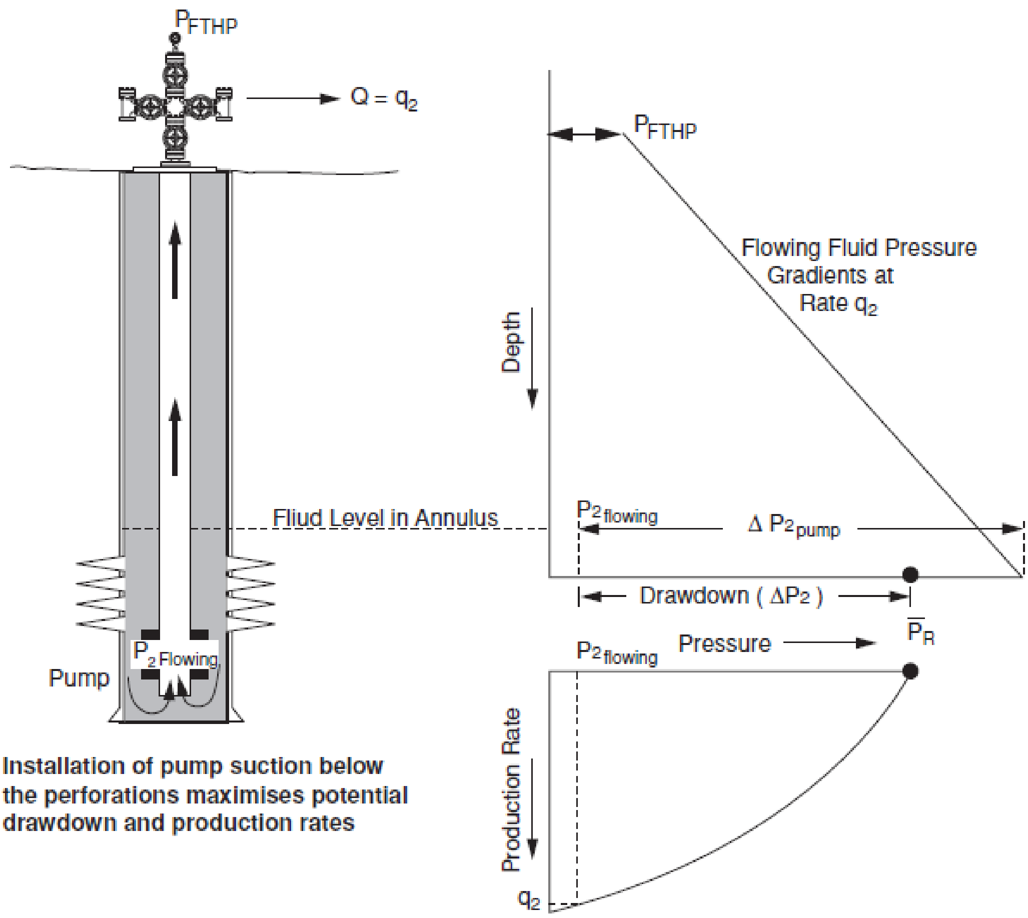


Figure 3c: Installation of pump below the perforations maximizes potential drawdown and production rates

The role of artificial lift in field development

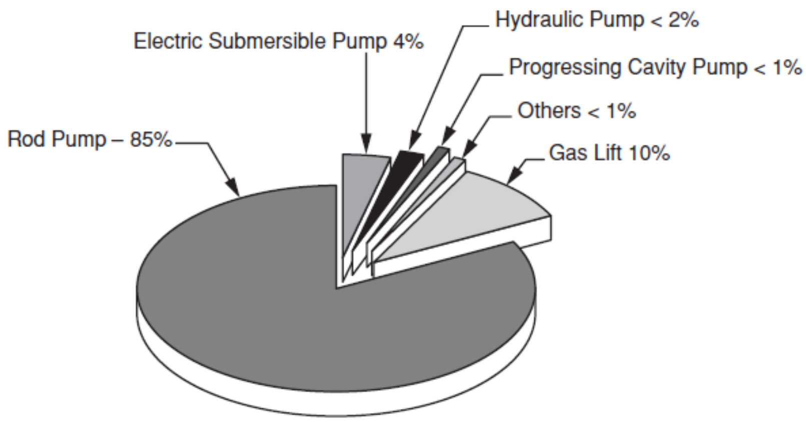


Figure 3.d: Artificial-lift methods installed in the USA in 1992 (J.Clegg, S.Buchan, N.Heln, JPT, December 1993)

The predominance of rod pumps indicates that the majority of wells are on land locations in mature fields and produce low rates per well.

Artificial lift is being more widely applied in field development than ever before due to:

- *Field development status*. Oil producing provinces such as the North Sea have become mature with the consequent reductions in bottom hole flowing pressure due to the depletion and increasing water cuts.

- *Absence of Pressure maintenance.* The development plans for many of the early giant North Sea fields employed early water injection to maintain the reservoir pressure above the hydrocarbon fluid's bubble point, which happened to be well above hydrostatic pressure. This meant that the high water cut wells still continued to flow at high production rates. Many of the current crops of smaller fields currently being developed do not employ any form of pressure maintenance resulting in an early need for artificial lift.

- *Satellite or Subsea Wells.* These wells are often positioned in a far distance from the host platform. The extra pressure drop caused by flow through these long, subsea pipelines needs to be overcome by some form of pressure boosting. This could either be an increased pressure boost from an ESP installed down hole or by a multiphase pump mounted on the sea bed.

- *Business drivers.* Profitable field development requires that the average well production rate exceed a minimum value with higher values being more profitable.

Well design can increase the well flow rate of return. Recent well design innovations include:

- Advanced well design: drilling of long horizontal exposures to the producing formation.
- Large diameter tubing to decrease the frictional pressure losses.
- Early installation of artificial lift to increase the flow rate.

Technical innovation

Technical innovation has increased the perspective of artificial lift. The development of multiphase pumps which are now available for both subsea and down hole application, new hybrid technologies such as down hole separation and pressure boosting for oil production to surface and water injection have made a big change in the reliability of the artificial lift equipment.

This can be achieved through:

- Improved engineering design
- The ability to monitor down hole conditions from the surface
- Better materials selection
- Better training of well site personnel who install and operate the equipment

Artificial lift selection criteria

The selection of proper artificial lift method is of a great importance and there are many factors that influence that choice.

Main factors to be considered are:

1. *Well and Reservoir Characteristics*

- Production casing size
- Maximum size of production tubing and required production rates
- Annular and tubing safety systems
- Producing formation depth and deviation
- Nature of the produced fluids (gas fraction and sand/wax/asphaltene production)
- Well inflow characteristics. A "straight line" inflow performance relationship associated with a dead oil is more favorable than the curved "Vogel" relationship when well inflow takes place below the fluid's bubble point.

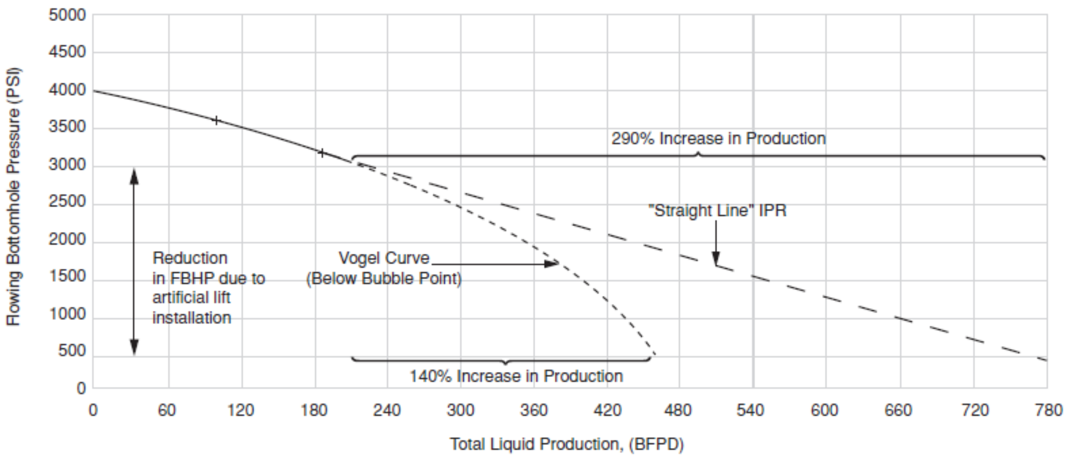


Figure 3.e: Influence of fluid inflow performance on the production increase achieved when well drawdown is increased

2. *Field Location*

- Offshore platform pursue the minimum physical size and weight of artificial lift equipment that can be installed.
- The on-shore environment can strongly influence the artificial lift selection. Different situations can lead to different artificial lift types being selected for wells of similar design and producing characteristics.
- Urban locations require the minimum of visual and acoustic impact
- A remote location with minimal availability of support infrastructure or regular access to the well
- Climatic extremes such as arctic operations will also limit the practical choices
- The distance from the wellhead to the processing facilities will also determine the minimum wellhead flowing pressure, which is required for a give production rate
- The available power source (natural gas, electricity, diesel generator) for the prime mover will impact the detailed equipment design and may effect reliability

3. *Operational Problems*

Some forms of artificial lift like gas lift are intrinsically more tolerant to solids production than centrifugal pumps. The formation of massive organic and inorganic deposits such as paraffins, asphaltenes, inorganic scales and hydrates will cause operational problems to the pumps and prevention by treatment with suitable inhibitors must take place.

The choice of materials used to manufacture the equipment installed within the well will depend on the:

- Bottom Hole Temperatures
- Corrosive Conditions e.g. H_2S , CO_2 , composition of the formation water
- Extent of Solids Production (erosion)
- Producing Velocities (erosion/corrosion)

4. *Economics*

A full life cycle economic analysis should carried out as high capital investment required to install artificial lift and the operating costs are normally much more important than the capital cost. It is viable to invest the necessary amount of money to ensure the most appropriate equipment is installed in the well if this will result in increased revenue or will reduce the operating costs.

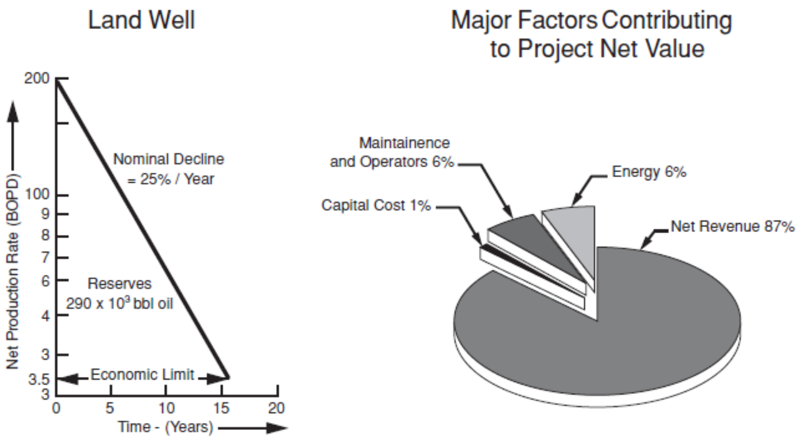


Figure 3.f: Full life cycle economics includes maintenances and other operational costs as well as capital costs

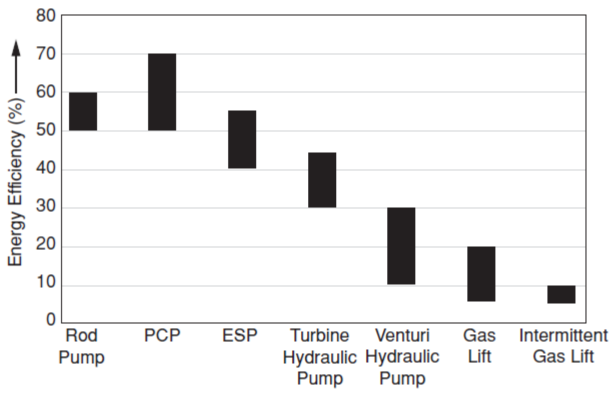


Figure 3.g: Comparison of the energy efficiency of the major artificial-lift methods adapted from J.Clegg et al (JPT, December 1993, p1128)

Maintenance costs will vary between operating locations depending on the state of the local, service company infrastructure and it can be costly in remote locations.

- The number of wells in the field with that particular form of artificial lift (economy of scale) will influence the operating costs.
- Similarly, the desirability and/or need for automation (how many operators are to be employed) and the decision as to whether or not to install centralized facilities will also influence the operating costs.
- The speed with which the "learning curve" is climbed for the more sophisticated forms of artificial lift will depend on the training provided and the skill base of the operations staff.

Implementation of Artificial lifts Selection Techniques

The artificial lift design engineer must determine the optimum type of artificial lift for a given field by matching facility constraints, artificial lift capabilities and the well productivity. Sometimes the type of lift has already been determined and the engineer has the problem of applying that system to the particular well.
There are certain environmental and geographical considerations that may be overriding. Sucker rod pumping may be eliminated as a suitable form of artificial lift if production is

required from the middle of a densely populated city or on an offshore platform with its limited deck area.
There are also practical limitations; deep wells producing several thousands of barrels per day that cannot be lifted by rod pumps. Thus, geographic and environmental considerations may make the decision.
Some types of artificial lift are able to reduce the sand face producing pressure to a very low value. The characteristics of the reservoir fluids must also be considered. Wax & formation solids present greater difficulties to some forms of artificial lift than others. The producing gas liquid ratio is key parameter to be considered by the artificial lift designer as gas represents a significant problem to all of the pumping methods.

Long Term Reservoir Performance and Facility Constraints

Another major factor that needs to be considered is long term reservoir performance. Two approaches both of which have disadvantages are used to solve the problem of artificial lift selection and sizing:

- A prediction of long term reservoir performance is made and artificial lift equipment installed that can handle the well’s production and producing conditions over its entire life.
  This choice leads to the installation of oversized equipment in the anticipation of ultimately producing large quantities of water. As a result, the equipment may have operated at poor efficiency due to under-loading over the early portion of its total life.

- The other extreme is to design for today's well producing conditions and not to worry about the future. This can lead to many changes in the type of lift equipment installed during the well’s producing life.
  Low cost operation may result in the short term but large sums of money will have to be spent later on to change the artificial lift equipment or the completion.

The selection of the artificial lift for a particular well must meet the physical constraints of the well. Once a particular type of lift is selected for use, consideration should be given to the size of the well bore required to obtain the desired production rate. Even if production rates can be achieved, smaller casing sizes can lead to higher, long term production costs due to well servicing problems or gas separation problems.
The correct way to design a well is to obtain an estimate of the expected production rates at various times in the field's life. The required size of the production tubing is estimated to allow these volumes of fluid to be produced.

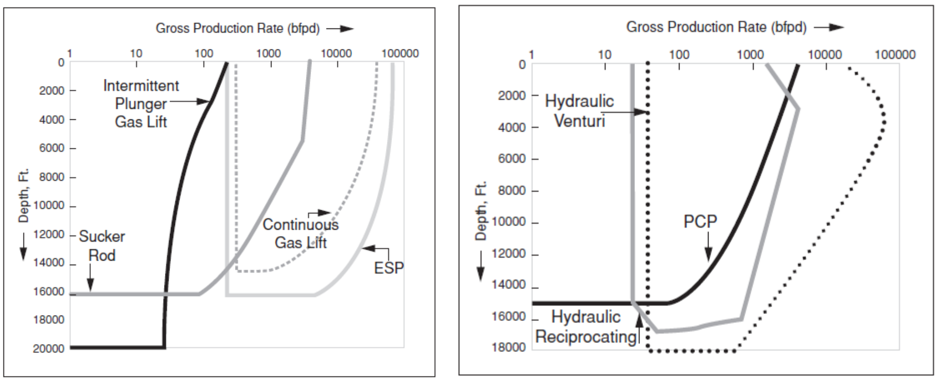


Figure 3.h: Typical application areas of artificial lift techniques

Design procedure steps

For any well using a down hole pump, the work supplied by the pump is related to the increase in pressure across the pump by the mechanical energy balance equation which for an incompressible fluid is:

$$W_s = \frac{p_2 - p_1}{\rho} + \frac{u_2^2 - u_1^2}{2g_c} + F$$

For liquids the kinetic energy term is usually small compared to the other terms so the equation is simplified to:

$$W_s = \frac{p_2 - p_1}{\rho} + F$$

where
W_s: is the work supplied by the pump
p_2: is the pressure in the tubing just above the pump
p_1: is the pressure just below the pump
F: is the frictional loss in the pump

To determine the size and power requirements for a down hole pump, the pressure on either side of the pump are related to the bottom hole flowing pressure by the pressure gradient in the gas-liquid stream below the pump and to the surface pressure by the single phase liquid gradient in the tubing.
A design procedure starts with the IPR relationship in which the needed p_{wf} for a desired production rate has been determined from a two-phase flow calculation. The pressure just below the pump p_1 is calculated from p_{wf} when the pump is near the production interval and $p_1 \approx p_{wf}$. From the surface tubing pressure p_2 is determined based on the single-phase liquid flow at the desired rate. Once the pressure increase required from the pump is known, the work required from the pump can be found usually based on empirical knowledge of the frictional losses in the pump.

3.1 Positive Displacement Pumps

3.1.1 Sucker Rod Pump

Sucker rod or beam pump was the first type of artificial lift to provide mechanical energy to lift oil from bottom hole to surface. It is the most widely used pump installations worldwide for reservoir production.
The low cost, mechanical simplicity and the ease with which efficient operation can be achieved makes rod pumps suitable for low, as 10 barrels of oil per day, volume operations. Rod pumps can lift moderate volumes (1,000 bfpd) from shallow depths (7,000 ft) or small volumes (200 bfpd) from greater depths (14,000 ft).

3.1.1.1 Pumping Unit

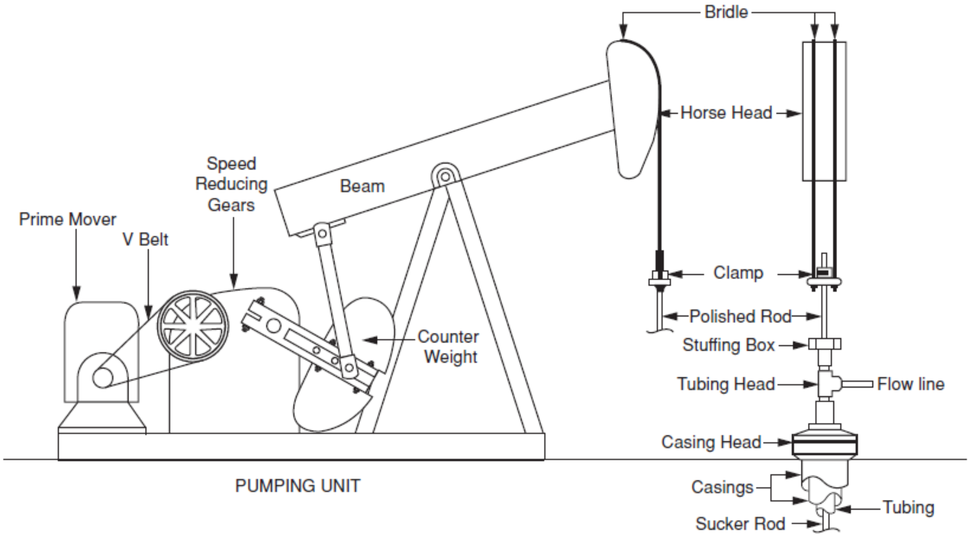


Figure 3.1.1.1.a: The surface equipment for a rod pump

The prime mover is normally an electric motor or gas engine and rotates at 600 revolutions per minute. It drives a set of speed reducing gears that reduces the speed to 20 strokes per minute or less.

The trend is to supply each well with its own motor, with and electric motors being more favorable because they can easily be automated. The power from the prime mover is transmitted to the input shaft of a gear reducer by a V-belt drive. The output shaft of the gear reducer drives the crank arm at a lower speed (4–40 revolutions per minute depending on well characteristics and fluid properties). The rotary motion of the crank arm is converted to an oscillatory motion by means of the walking beam through a pitman arm. The horse’s head and the hanger cable arrangement is used to ensure that the upward pull on the sucker rod string is vertical at all times.

Conventional pumping units are available in a wide range of sizes, with stroke lengths varying from 12 to almost 200 in. The strokes for any pumping unit type are available in increments. The stroke length can be varied within limits; about six different lengths are possible. These different lengths are achieved by varying the position of the pitman arm connection on the crank arm. Walking beam ratings are expressed in allowable polished rod loads (PRLs) and vary from approximately 3,000 to 35,000 lb.

Counterbalance for conventional pumping units is accomplished by placing weights directly on the beam or by attaching weights to the rotating crank arm for smaller units. A combination of the two methods is being used for larger units. In more recent designs, the rotary counterbalance can be adjusted by shifting the position of the weight on the crank by a jackscrew or rack and pinion mechanism.

There are two other major types of pumping units except of the conventional. These are the *Lufkin Mark II* and the *Air-Balanced Units*. The pitman arm and horse’s head are in the same side of the walking beam in both types of these units. Air cylinders are used in the air-balanced units to balance the torque on the crankshaft, instead of counter-weights *in Lufkin Mark II* unit.

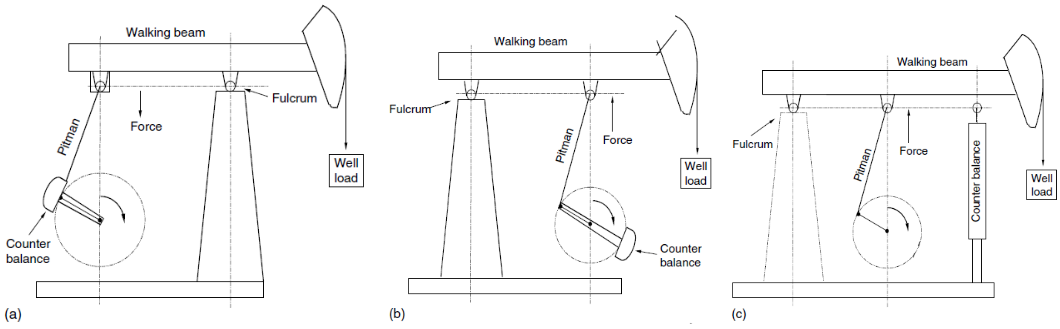


Figure 3.1.1.1.b: Sketch of three types of pumping units: (a) conventional unit; (b) Lufkin Mark II Unit; (c) air-balanced unit

The connection between the surface pumping unit and the down hole pump is the polished rod and the sucker rods. The polished rod moves up and down through a stuffing box mounted on top of the wellhead. This stuffing box seals against the polished rod and prevents surface leaks of the liquid and gas being produced by the well.

Designations for sucker rod pumping units

The *American Petroleum Institute* (API) has established designations for sucker rod pumping units using a string of characters containing four fields. For example:

C-228D-200-74

The first field is the code for type of pumping unit. C is for conventional units, A is for air-balanced units, B is for beam counterbalance units, and M is for Mark II units. The second field is the code for peak torque rating in thousands of inch-pounds and gear reducer. D stands for double-reduction gear reducer. The third field is the code for PRL rating in hundreds of pounds. The last field is the code for stroke length in inches.

3.1.1.2 The Sucker Rods

The sucker rods, typically 25 ft long, are circular steel rods with diameters between 0.5 in and 1.125 in. A threaded male connection or pin is machined at each end of the rod. The two rods can be joined together by use of a double box coupling (Figure 3.1.1.2.a). Square flats are machined near the pins and at the center of the coupling to provide a grip for a wrench to allow the rods and couplings to be screwed together.

The sucker rods are subjected to continuous fatigue when the pump is operating. The weight of the rod string (one component of this fatigue load) can be minimized by using a tapered sucker rod string. This involves installing lighter, smaller diameter rods lower down in the well where the load they have to support (weight of rods and fluid in the tubing string) is less than at the top of the well. Glass fiber rods and continuous solid rods are also in use.

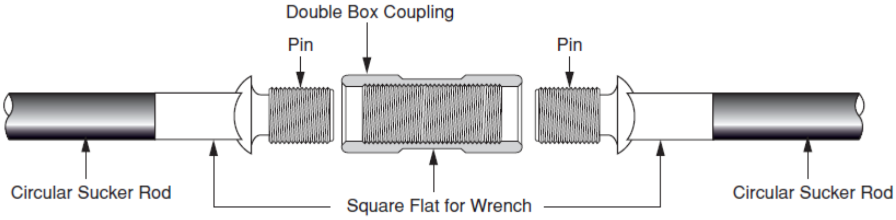


Figure 3.1.1.2.a: Sucker rods are joined together by a coupling

The Pump

The pump is installed in the tubing string below the dynamic liquid level. It consists of a working barrel and liner, standing valve (SV), and traveling valve (TV) at the bottom of the plunger, which is connected to sucker rods. As the plunger is moved downward by the sucker rod string, the TV is open, which allows the fluid to pass through the valve, which lets the plunger move to a position just above the SV. During this downward motion of the plunger, the SV is closed; thus, the fluid is forced to pass through the TV. When the plunger is at the bottom of the stroke and starts an upward stroke, the TV closes and the SV opens. As upward motion continues, the fluid in the well below the SV is drawn into the volume above the SV (fluid passing through the open SV). The fluid continues to fill the volume above the SV until the plunger reaches the top of its stroke.

The pump is located near the perforations at the bottom of the sucker rod string. It consists of a hollow plunger with circular sealing rings mounted on the outside circumference moving inside a pump barrel which is either inserted into the tubing or is part of the tubing itself. The Standing valve is mounted at the bottom of the pump barrel while the Travelling valve is installed at the top of the plunger. The Standing and Travelling valves contain of a ball which closes the passage in the pump inlet and plunger when the ball is seated; they act non-return valves.

The “UP” and “DOWN” movement of the pump barrel allows the fluid pressure in the pump barrel to open and shut the Travelling and Standing valves. The "Upward" rod movement reduces the pressure within the pump barrel and the upward flow of fluid from the perforations below the pump lifts the Standing valve’s ball off its seat. The pressure due to the fluid column above the plunger keeps the Travelling valve ball on its seat. The situation is reversed during the “DOWN” stroke – compression of fluid within the pump barrel forces it to flow through the hollow plunger and to lift the Travelling valve off its seat; while ensuring that the Standing valve remains closed.

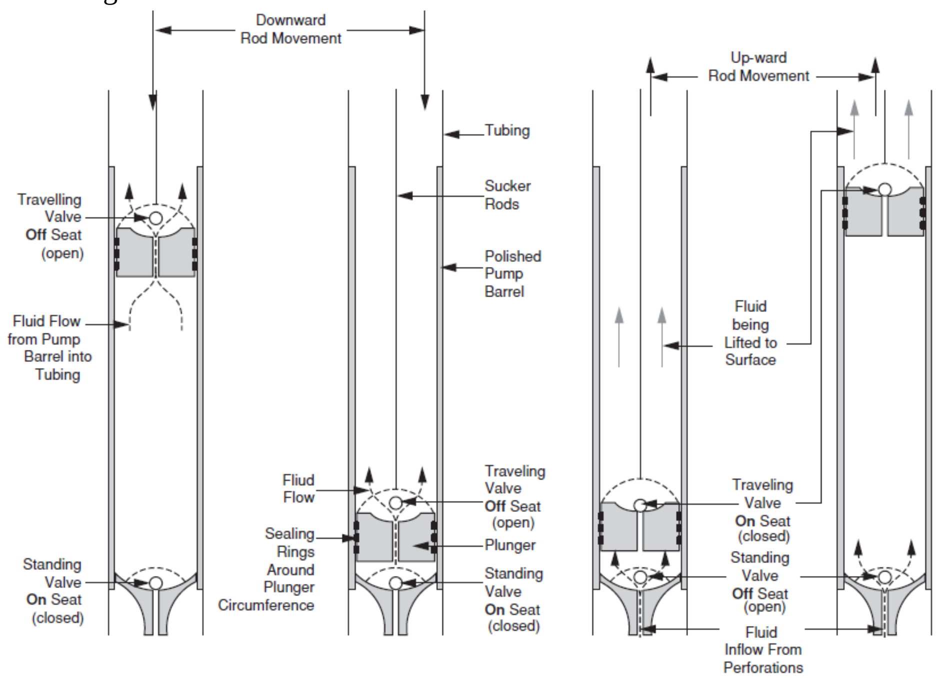


Figure 3.1.1.2.b: Rod pump operation

There are two basic types of plunger pumps: tubing pump and rod pump (Fig. 3.1.1.2.c). For the tubing pump, the working barrel or liner (with the SV) is attached to the bottom of the production tubing string and must be run into the well with the tubing. The plunger (with the TV) is inside the tubing on the sucker rod string. Once the plunger is seated in the working barrel, pumping can be initiated. For the rod pump both working barrel and plunger are run into the well on the sucker rod string and is seated on a wedged type seat that is fixed to the bottom joint of the production tubing. Plunger diameters vary from $^5/_8$ to 4 $^5/_8$ in.

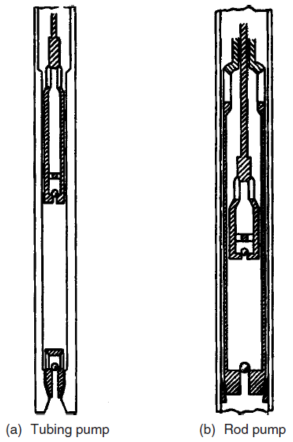


Figure 3.1.1.2.c: Two types of plunger pumps (Nind, 1964)

3.1.1.3 Polished Rod Motion

Figure 3.1.1.3.a shows the cyclic motion of a polished rod in its movements through the stuffing box of the conventional pumping unit and the air-balanced pumping unit.

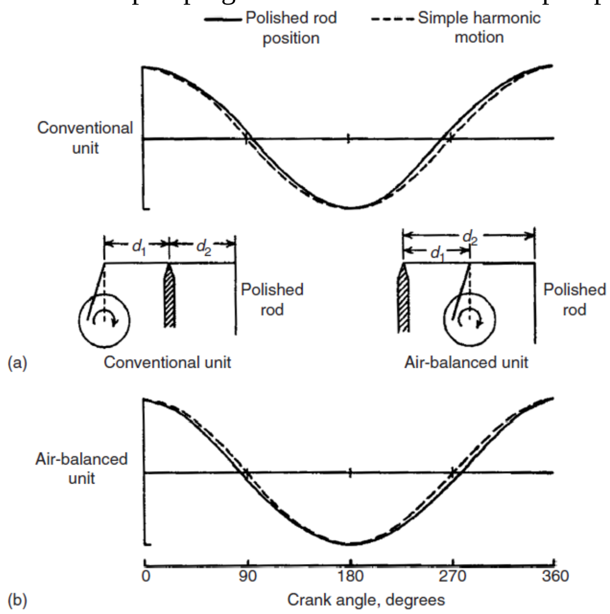


Figure 3.1.1.3.a: Polished rod motion for (a) conventional pumping unit and (b) air-balanced unit (Nind, 1964)

For Conventional Pumping Unit the acceleration at the bottom of the stroke 180°-360° is somewhat greater than true simple harmonic acceleration. At the top of the stroke 180°, it is less. This is a major drawback for the conventional unit. Just at the time the TV is closing and the fluid load is being transferred to the rods, the acceleration for the rods is at its maximum. These two factors combine to create a maximum stress on the rods that becomes one of the limiting factors in designing an installation.

For Air-Balanced Pumping Unit the maximum acceleration occurs at the top of the stroke 360° (the acceleration at the bottom of the stroke is less than simple harmonic motion 180°). Thus, a lower maximum stress is set up in the rod system during transfer of the fluid load to the rods.

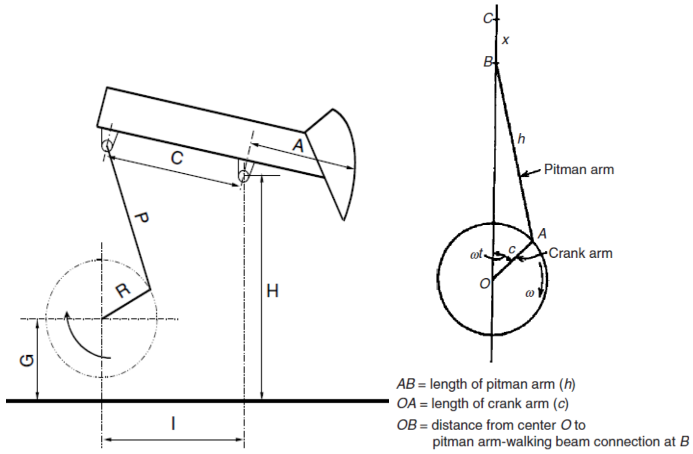


Figure 3.1.1.3.b: Definitions of conventional pumping unit API geometry dimensions and approximate motion of connection point between pitman arm and walking beam (Nind, 1964)

The following analyses of polished rod motion apply to conventional units.
If x denotes the distance of B below its top position C and is measured from the instant at which the crank arm and pitman arm are in the vertical position with the crank arm vertically upward, the law of cosine gives:

$(AB)^2 = (OA)^2 + (OB)^2$ - 2(OA)(OB) $\cos AOB$,
that is, $(h)^2 = (c)^2 + (h + c - x)^2 - 2c(h + c - x) \cos \omega t$,

where ω is the angular velocity of the crank. The equation reduces to

$$x^2 - 2x[h + c(1 - cos\omega t)] + 2c(h + c)(1 - cos\omega t) = 0$$

so that $x = h + c(1 - cos\omega t) \pm \sqrt{c^2 cos^2 \omega t + (h^2 - c^2})$

When ωt is zero, x is also zero, which means that the negative root sign must be taken.

Therefore, $x = h + c(1 - cos\omega t) - \sqrt{c^2 cos^2 \omega t + (h^2 - c^2}$

Acceleration is

$$a = \frac{d^2 x}{dt^2}$$

Carrying out the differentiation for acceleration, it is found that the maximum acceleration occurs when ωt is equal to zero (or an even multiple of p radians) and that this maximum value is

$$a_{max} = \omega^2 c(1 + \frac{c}{h}) \qquad (3.1)$$

It also appears that the minimum value of acceleration is

$$a_{max} = \omega^2 c(1 - \frac{c}{h}) \qquad (3.2)$$

If N is the number of pumping strokes per minute, then

$$\omega = \frac{2\pi N}{60} \ ({rad}/_{sec}) \qquad (3.3)$$

The maximum downward acceleration of point B (which occurs when the crank arm is vertically upward) is

$$a_{max} = \frac{cN^2}{91.2}\left(1 + \frac{c}{h}\right) \ ({ft}/_{sec^2}) \qquad (3.4) \qquad \text{or} \qquad a_{max} = \frac{cN^2 g}{2936.3}\left(1 + \frac{c}{h}\right) \ ({ft}/_{sec^2}) \qquad (3.5)$$

Likewise the minimum upward (a_{min}) acceleration of point B (which occurs when the crank arm is vertically downward) is

$$a_{min} = \frac{cN^2 g}{2936.3}\left(1 - \frac{c}{h}\right) \ ({ft}/_{sec^2}) \qquad (3.6)$$

It follows that in a conventional pumping unit, the maximum upward acceleration of the horse's head occurs at the bottom of the stroke (polished rod) and is equal to

$$a_{max} = \frac{d_1}{d_2}\frac{cN^2 g}{2936.3}\left(1 + \frac{c}{h}\right) \ ({ft}/_{sec^2}) \qquad (3.7)$$

where d_1 and d_2 are shown in Fig. 3.1.1.3.a.

However $\frac{2cd_2}{d_1} = S,$ where S is the polished rod stroke length.

So if S is measured in inches, then

$$\frac{2cd_2}{d_1} = \frac{S}{12} \qquad \text{or} \qquad \frac{cd_2}{d_1} = \frac{S}{24} \qquad (3.8)$$

by substitution Eq.(3.8) into Eq.(3.7)

$$a_{max} = \frac{SN^2 g}{70471.2}\left(1 + \frac{c}{h}\right) \ ({ft}/_{sec^2}) \qquad (3.9)$$

or

$$a_{max} = \frac{SN^2 g}{70471.2} M \ ({ft}/_{sec^2}) \qquad (3.10)$$

Where M is the machinery factor and is defined as M=1 $+ \frac{c}{h}$ (3.11)

for the air-balanced unit, M is replaced by 1-c/h

Similarly

$$a_{min} = \frac{SN^2 g}{70471.2}\left(1 - \frac{c}{h}\right) \ ({ft}/_{sec^2}) \qquad (3.12)$$

For air-balanced units, because of the arrangements of the levers, the acceleration defined in Eq. (3.12) occurs at the bottom of the stroke, and the acceleration defined in Eq. (3.9) occurs at the top. With the lever system of an air-balanced unit, the polished rod is at the top of its stroke when the crank arm is vertically upward.

3.1.1.4 Pumping Limit Load Calculations

The maximum and minimum polished rod loads, the peak torque and the theoretical horsepower required can be calculated once the pump speed, stroke length, plunger diameter and rod sizes have been chosen.

Load to the Pumping Unit

The load exerted to the pumping unit depends on well depth, rod size, fluid properties, and system dynamics. The maximum PRL and peak torque are major concerns for pumping unit.

Maximum PRL

The PRL is the sum of weight of fluid being lifted, weight of plunger, weight of sucker rods string, dynamic load due to acceleration of the rods, friction force and the up-thrust from below on plunger. The friction term and the weight of the plunger are neglected. TV is assumed to be closed at the instant at which the acceleration term reaches its maximum. With these assumptions, the PRL_{max} becomes

$$PRL_{max} = S_f(62.4)D\frac{(A_p - A_r)}{144} + \frac{\gamma_s D A_r}{144} + \frac{\gamma_s D A_r}{144}\left(\frac{SN^2M}{70{,}471.2}\right) \quad (3.13)$$

Where:

S_f = Specific gravity of fluid in tubing

D = length of sucker rod string (ft)

A_p=Gross plunger cross-sectional area (in^2)

A_r= Sucker rod cross-sectional area (in^2)

γ_s = Specific weight of steel ($490\ lb/ft^3$)

Equation (3.13) can be rewritten as:

$$PRL_{max} = S_f(62.4)D\frac{A_p}{144} - S_f(62.4)D\frac{A_r}{144} + \frac{\gamma_s D A_r}{144} + \frac{\gamma_s D A_r}{144}\left(\frac{SN^2M}{70{,}471.2}\right) \quad (3.14)$$

If the weight of the rod string in air is

$$W_r = \frac{\gamma_s D A_r}{144} \quad (3.15)$$

which can be solved for A_r, which is

$$A_r = \frac{144 W_r}{\gamma_s D} \quad (3.16)$$

Substituting Eq. (3.16) into Eq. (3.14) yields

$$PRL_{max} = S_f(62.4)D\frac{A_p}{144} - S_f(62.4)\frac{W_r}{\gamma_s} + W_r + W_r\left(\frac{SN^2M}{70{,}471.2}\right) \quad (3.17)$$

The above equation is often further reduced by taking the fluid in the second term (the subtractive term) as a 50 8API with S_f = 0.78. Thus, Eq. (3.17) becomes (where γ_s= 490)

$$PRL_{max} = S_f(62.4)D\frac{A_p}{144} - 0.1W_r + W_r + W_r\left(\frac{SN^2M}{70{,}471.2}\right)$$

or

$$PRL_{max} = W_f + 0.9W_r + W_r\left(\frac{SN^2M}{70{,}471.2}\right) \quad (3.18)$$

where $W_f = S_f(62.4)D\frac{A_p}{144}$ and is called the fluid load (not to be confused with the actual fluid weight on the rod string).

Thus, Eq. (3.18) can be rewritten as: $$PRL_{max} = W_f + (0.9 + F_1)W_r \qquad (3.19)$$
where for conventional units
$$F_1 = \frac{SN^2(1+\frac{c}{h})}{70{,}471.2} \qquad (3.20)$$
and for air-balanced units
$$F_1 = \frac{SN^2(1-\frac{c}{h})}{70{,}471.2} \qquad (3.21)$$

Minimum PRL

The minimum PRL occurs while the TV is open so that the fluid column weight is carried by the tubing and not the rods. The minimum load is at or near the top of the stroke. Neglecting the weight of the plunger and friction term, the minimum PRL is

$$PRL_{min} = -S_f(62.4)\frac{W_r}{\gamma_s} + W_r - W_rF_2$$

which, for 50 8API oil, reduces to: $$PRL_{min} = 0.9W_r - F_2W_r = (0.9 - F_2)W_r \qquad (3.22)$$

where for the conventional units: $$F_2 = \frac{SN^2(1-\frac{c}{h})}{70{,}471.2} \qquad (3.23)$$

and for air-balanced units: $$F_2 = \frac{SN^2(1+\frac{c}{h})}{70{,}471.2} \qquad (3.24)$$

Counterweights

To reduce the power requirements for the prime mover, a counterbalance load is used on the walking beam (small units) or the rotary crank. The ideal counterbalance load C is the average PRL. Therefore: $C = \frac{1}{2}(PRL_{max} + PRL_{min})$

Using Eqs. (3.19) and (3.22) we get: $$C = \frac{1}{2}\ W_f\ +\ 0.9\ W_r + \frac{1}{2}(F_1 - F_2)W_r \qquad (3.25)$$

or for conventional units: $$C = \frac{1}{2}\ W_f\ +\ W_r + (0.9\ +\ \frac{SN^2}{70{,}471.2}\ \frac{c}{h}) \qquad (3.26)$$

and for air-balanced units: $$C = \frac{1}{2}\ W_f\ +\ W_r + (0.9 -\ \frac{SN^2}{70{,}471.2}\ \frac{c}{h}) \qquad (3.27)$$

The counterweights can be selected from manufacturer's catalog based on the calculated C value. The relationship between the counterbalance load C and the total weight of the counterweights is:

$$C = C_s + W_c(\frac{r}{c}\frac{d_1}{d_2})$$

where
C_s = Structure unbalance, lb
W_c = Total weight of counterweights, lb
r = distance between the mass center of counterweights and the crank shaft center, in.

Peak Torque and Speed Limit

The peak torque exerted is usually calculated on the most severe possible assumption, which is that the peak load (polished rod less counterbalance) occurs when the effective crank length is also a maximum (when the crank arm is horizontal). Thus, peak torque T is:

$$T = c[C - (0.9 -\ F_{2)}W_r]\frac{d_2}{d_1} \qquad (3.28)$$

Substituting Eq. (3.25) into Eq. (3.28) gives: $T = \frac{1}{2}S[C - (0.9 - F_{2)}W_r]$ (3.29)

or $T = \frac{1}{2}S[\frac{1}{2}W_f + \frac{1}{2}(F_1 + F_{2)}W_r]$ or $T = \frac{1}{4}S(W_f + \frac{2SN^2W_r}{70,471.2})$ (in-lb) (3.30)

Because the pumping unit itself is usually not perfectly balanced ($C_s \neq 0$), the peak torque is also affected by structure unbalance. Torque factors are used for correction:

$$T = \frac{\frac{1}{2}[PRL_{\max}(TF_1) + PRL_{min}(TF_2)]}{0.93} \tag{3.31}$$

where
TF_1 = Maximum upstroke torque factor
TF_2 = Maximum downstroke torque factor
0.93 = system efficiency

For symmetrical conventional and air-balanced units, $TF = TF_1 = TF_2$
There is a limiting relationship between stroke length and cycles per minute. As given earlier, the maximum value of the downward acceleration (which occurs at the top of the stroke) is equal to:

$$a_{max/min} = \frac{SN^2g(1 \pm \frac{c}{h})}{70,471.2} \tag{3.32}$$

If this maximum acceleration divided by g exceeds unity, the downward acceleration of the hanger is greater than the free-fall acceleration of the rods at the top of the stroke. This leads to severe pounding when the polished rod shoulder falls onto the hanger (leading to failure of the rod at the shoulder). Thus, a limit of the above downward acceleration term divided by g is limited to approximately 0.5 (or where L is determined by experience in a particular field). Thus,

$$\frac{SN^2g\left(1 \pm \frac{c}{h}\right)}{70,471.2} \leq L \tag{3.33}$$

or

$$N_{limit} = \sqrt{\frac{70,471.2\, L}{S(1 \mp \frac{c}{h})}} \tag{3.34}$$

for L=0.5

$$N_{limit} = \frac{187.7}{\sqrt{S(1 \mp \frac{c}{h})}} \tag{3.35}$$

The minus sign is for conventional units and the plus sign is for air-balanced units.

Tapered Rod Strings

For deep well applications it is necessary to use a tapered sucker rod strings to reduce the PRL at the surface. The larger diameter rod is placed at the top of the rod string, then the next largest, and then the least large. Usually these are in sequences up to four different rod sizes and tapered rod strings are designated by 1/8-in. increments. Tapered rod strings can be identified by their numbers such as:

a) No. 88 is a non-tapered 8⁄8 - or 1-in. diameter rod string
b) No. 76 is a tapered string with 7⁄8 -in. diameter rod at the top, then a 6⁄8 -in. diameter rod at the bottom.
c) No. 75 is a three-way tapered string consisting of 7⁄8 -in. diameter rod at top 6⁄8 -in. diameter rod at middle 5⁄8 -in. diameter rod at bottom

d) No. 107 is a four-way tapered string consisting of 10⁄8 -in. diameter rod at top 9⁄8 -in. 8⁄8 -in. and 7⁄8 –in. at the end.

Tapered rod strings are designed for static loads with a sufficient factor of safety to allow for random low level dynamic loads. Two criteria are used in the design of tapered rod strings:

1. Stress at the top rod of each rod size is the same throughout the string.
2. Stress in the top rod of the smallest (deepest) set of rods should be the highest (30,000 psi) and the stress progressively decreases in the top rods of the higher sets of rods.

The reason for the second criterion is that it is preferable that any rod breaks occur near the bottom of the string.

Effective Plunger Stroke Length

The motion of the plunger at the pump-setting depth does not coincide with the motion of the polished rod in time and magnitude because sucker rods and tubing strings are elastic. Plunger motion depends on: polished rod motion, sucker rod stretch and tubing stretch.

Two major sources of difference in the motion of the polished rod and the plunger are elongation of the rod string and over travel. Stretch is caused by the periodic transfer of the fluid load from the SV to the TV and back again. The result is a function of the stretch of the rod string and the tubing string. Rod string stretch is caused by the weight of the fluid column in the tubing coming on to the rod string at the bottom of the stroke when the TV closes. It is apparent that the plunger stroke will be less than the polished rod stroke length S by an amount equal to the rod stretch. The magnitude of the rod stretch is

$$\delta l_r = \frac{W_f D_r}{A_r E} \tag{3.36}$$

where

W_f = weight of fluid (lb)

D_r = length of rod string (ft)

A_r = cross-sectional area of rods (in^2)

E = modulus of elasticity of steel (30x10^6 $lb/_{in^2}$)

Tubing stretch can be expressed by a similar equation: $\delta l_t = \frac{W_f D_t}{A_t E}$ (3.37)

But because the tubing cross-sectional area A_t is greater than the rod cross-sectional area A_r, the stretch of the tubing is small and is usually neglected. However, the tubing stretch can cause problems with wear on the casing. Thus, for this reason a tubing anchor is almost always used. Plunger overt ravel at the bottom of the stroke is a result of the upward acceleration imposed on the downward moving sucker rod elastic system. An approximation to the extent of the over travel may be obtained by considering a sucker rod string being accelerated vertically upward at a rate n times the acceleration of gravity. The vertical force required to supply this acceleration is nW_r. The magnitude of the rod stretch due to this force is:

$$\delta l_o = n \frac{W_r D_r}{A_r E} \ (ft) \tag{3.38}$$

But the maximum acceleration term n can be written as

$$\frac{SN^2\left(1 \pm \frac{c}{h}\right)}{70{,}471.2}$$

so that Eq. (3.38) becomes

$$\delta l_o = \frac{SN^2\left(1 \pm \frac{c}{h}\right)}{70{,}471.2} \frac{W_r D_r}{A_r E}\ (ft) \quad (3.39)$$

where the plus sign applies to conventional units and the minus sign to air-balanced units. For conventional units Eq. (3.39) becomes:

$$\delta l_o = \frac{W_r D_r}{A_r E} \frac{SN^2 M}{70{,}471.2}\ (ft) \quad (3.40)$$

Equation (3.40) can be rewritten to yield δl_o in inches. $W_r = \gamma_s A_r D_r$ and $\gamma_s = 490\ {}^{lb}/_{ft^3}$ with E= 30x10^6 ${}^{lb}/_{ft^2}$

Eq. (3.40) becomes: $\delta l_o = 1.93x10^{-11} D_r^2 SN^2 M$ (in) (3.41)

Plunger stroke is approximated using the above expressions as: $S_p = S - \delta l_r - \delta l_t + \delta l_o$

or $$S_p = S - \frac{12D}{E}\left[W_f\left(\frac{1}{A_r} + \frac{1}{A_t}\right) - \frac{SN^2 M}{70{,}471.2} \frac{W_r}{A_r}\right] \quad (3.42)$$

If pumping is carried out at the maximum permissible speed limited by Eq. (3.34), the plunger stroke becomes:

$$S_p = S - \frac{12D}{E}\left[W_f\left(\frac{1}{A_r} + \frac{1}{A_t}\right) - \frac{1+\frac{c}{h}}{1-\frac{c}{h}} \frac{LW_r}{A_r}\right] \quad (3.43)$$

For the air-balanced unit, the term $\frac{1+\frac{c}{h}}{1-\frac{c}{h}}$ is replaced by its reciprocal.

Pump Rate

The pump rate Q is related to the volume displaced V by each pump stroke and the speed rate or number of strokes per minute N. Thus:

Q = K*V*N*f = K*A*S*N*f

where:

A: is the area of the pump barrel

S: is the length of the pump stroke

f: is the efficiency factor

K: is a constant depending on the units employed

The maximum speed N of the pump unit is determined by the speed at which the sucker rods fall downward in the “DOWN” stroke. This maximum speed decreases as the length of the pump stroke increases. Typical maximum values are quoted in Table 3.1.1.4:

Maximum Allowable Pump Speed †

Stroke Length (in.)	30	60	90	120	180	240	300
Maximum Pump Speed (SPM*)	34	24	19	17	14.5	11.5	10.5

* Strokes Per Minute

Table 3.1.1.4: Maximum allowable pump speed for a conventional pump unit

Low pump speeds and large diameter pumps lead to the greatest energy efficiency, but also the largest equipment loads. It is common practice to put a few larger rods capable of carrying any compression loads due to buckling at the bottom of the rod string. The addition of sinker bars will increase the rate of rod fall, but also increase the load on the rods.

Volumetric Efficiency

Volumetric efficiency of the plunger mainly depends on the rate of slippage of oil past the pump plunger and the solution–gas ratio under pump condition. Metal-to-metal plungers are commonly available with plunger-to-barrel clearance on the diameter of -0.001, -0.002,-0.003,-0.004, and -0.005 in.
Such fits are referred to as -1, -2, -3, -4, and -5, meaning the plunger outside diameter is 0.001 in. smaller than the barrel inside diameter. In selecting a plunger, one must consider the viscosity of the oil to be pumped. A loose fit may be acceptable for a well with high viscosity oil (low 8API gravity). But such a loose fit in a well with low viscosity oil may be very inefficient. Guidelines are as follows:

a. Low-viscosity oils (1–20 cps) can be pumped with a plunger to barrel fit of -0.001 in.
b. High-viscosity oils (7,400 cps) will probably carry sand in suspension so a plunger-to-barrel fit or approximately 0.005 in. can be used.

An empirical formula has been developed that can be used to calculate the slippage rate, q_s (bbl/day), through the annulus between the plunger and the barrel:

$$q_s = \frac{k_p}{\mu}\,\frac{\left(d_b - d_p\right)^{2.9}(d_b + d_p)}{{d_b}^{0.1}}\,\frac{\Delta_p}{L_p}$$

where
$k_p =$ a constant
$d_p =$ plunger outside diameter (in.)
$d_b =$ barrel inside diameter (in.)
$\Delta_p =$ differential pressure drop across plunger (psi)
$L_p =$ length of plunger (in.)
μ = viscosity of oil (cp)

The value of k_p is 2.77x10^6 to 6.36x10^6depending on field conditions. An average value is 4.17x10^6. The value of Δ_p may be estimated on the basis of well productivity index and production rate. A reasonable estimate may be a value that is twice the production drawdown.

Volumetric efficiency can decrease significantly due to the presence of free gas below the plunger. As the fluid is elevated and gas breaks out of solution, there is a significant difference between the volumetric displacement of the bottom-hole pump and the volume of the fluid delivered to the surface. The effect of gas on volumetric efficiency depends on solution–gas ratio and bottom-hole pressure. Down-hole devices, called ‘‘gas anchors,’’ are usually installed on pumps to separate the gas from the liquid. Pump efficiency can vary over a wide range but are commonly 70–80%.

Power Requirements

The prime mover should be properly sized to provide adequate power to lift the production fluid, to overcome friction loss in the pump, in the rod string and polished rod and in the pumping unit.
It is usually expressed in terms of net lift:

$$P_h = 7.36x10^{-6} q\gamma_l\, L_N$$

where

P_h = hydraulic power, hp
q = liquid production rate, bbl/day
γ_l = liquid specific gravity, water ¼ 1
L_N = net lift, ft,

$$L_N = H + \frac{p_{tf}}{0.433\gamma_l}$$

where
H = depth to the average fluid level in the annulus, ft
p_{tf} = flowing tubing head pressure, psig.

The power required to overcome friction losses can be empirically estimated as

$$P_f = 6.31x10^{-7}W_r\,SN$$

Thus, the required prime mover power can be expressed as

$$P_{pm} = F_s(P_h + P_f)$$

where F_s is a safety factor of 1.25–1.50.

3.1.1.5 Procedure for Pumping Unit Selection

The following procedure can be used for selecting a pumping unit:

1. From the maximum anticipated fluid production (based on IPR) and estimated volumetric efficiency, the required pump displacement can be calculated.
2. Based on well depth and pump displacement, the API rating and stroke length of the pumping unit to be used can be determined. This can be done using either Fig. 3.1.15 or Table 3.1.1.5.
3. Selection of tubing size, plunger size, rod sizes, and pumping speed is being made from Table 3.1.1.5.
4. The fractional length of each section of the rod string is being calculated
5. The length of each section of the rod string to the nearest 25 ft is being calculated
6. Calculation of the acceleration factor is needed.
7. The effective plunger stroke length is being determined.
8. Using the estimated volumetric efficiency, the probable production rate is determined and checked against the desired production rate.
9. Calculation of the dead weight of the rod string is necessary
10. The fluid load is calculated.
11. The peak polished rod load is determined and checked against the maximum beam load for the unit selected.
12. Calculation of the maximum stress at the top of each rod size is being done and checked against the maximum permissible working stress for the rods to be used.
13. Calculation of the ideal counterbalance effect is also needed and checked against the counterbalance available for the unit selected.
14. From the manufacturer's literature, the position of the counterweight to obtain the ideal counterbalance effect must be selected.
15. On the assumption that the unit will be no more than 5% out of counterbalance, the peak torque on the gear reducer must be calculated and checked against the API rating of the unit selected.
16. Calculation of hydraulic horsepower, friction horsepower and brake horsepower of the prime mover is necessary for the select of the prime mover.

17. From the manufacturer's literature, the gear reduction ratio, unit sheave size and the speed of the prime mover are obtained.

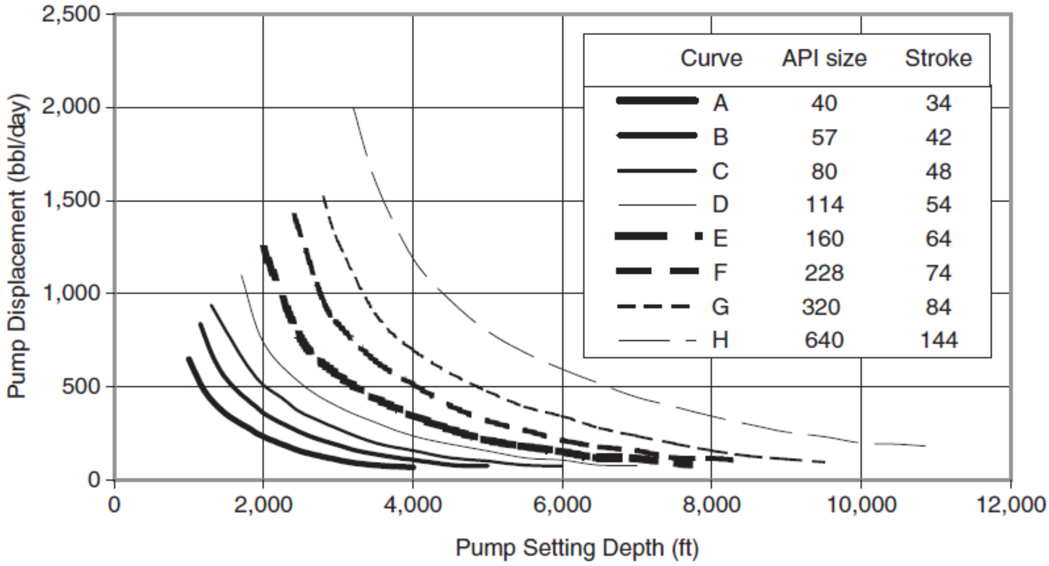


Figure 3.1.1.5: Sucker rod pumping unit selection chart (Kelley and Willis, 1954)

Design Data for API Sucker Rod Pumping Units

(a) Size 40 unit with 34-in. stroke

Pump depth (ft)	*Plunger size* (in.)	*Tubing size* (in.)	*Rod sizes* (in.)	*Pumping speed* (stroke/min)
1,000–1,100	$2\frac{3}{4}$	3	$\frac{7}{8}$	24–19
1,100–1,250	$2\frac{1}{2}$	3	$\frac{7}{8}$	24–19
1,250–1,650	$2\frac{1}{4}$	$2\frac{1}{2}$	$\frac{3}{4}$	24–19
1,650–1,900	2	$2\frac{1}{2}$	$\frac{3}{4}$	24–19
1,900–2,150	$1\frac{3}{4}$	$2\frac{1}{2}$	$\frac{3}{4}$	24–19
2,150–3,000	$1\frac{1}{2}$	2	$\frac{5}{8}$–$\frac{3}{4}$	24–19
3,000–3,700	$1\frac{1}{4}$	2	$\frac{5}{8}$–$\frac{3}{5}$	22–18
3,700–4,000	1	2	$\frac{5}{8}$–$\frac{3}{6}$	21–18

(b) Size 57 unit with 42-in. stroke

Pump depth (ft)	*Plunger size* (in.)	*Tubing size* (in.)	*Rod sizes* (in.)	*Pumping speed* (stroke/min)
1,150–1,300	$2\frac{3}{4}$	3	$\frac{7}{8}$	24–19
1,300–1,450	$2\frac{1}{2}$	3	$\frac{7}{8}$	24–19
1,450–1,850	$2\frac{1}{4}$	$2\frac{1}{2}$	$\frac{3}{4}$	24–19
1,850–2,200	2	$2\frac{1}{2}$	$\frac{3}{4}$	24–19
2,200–2,500	$1\frac{3}{4}$	$2\frac{1}{2}$	$\frac{3}{4}$	24–19
2,500–3,400	$1\frac{1}{2}$	2	$\frac{5}{8}$–$\frac{3}{4}$	23–18
3,400–4,200	$1\frac{1}{4}$	2	$\frac{5}{8}$–$\frac{3}{5}$	22–17
4,200–5,000	1	2	$\frac{5}{8}$–$\frac{3}{6}$	21–17

Table 3.1.1.5: Design data for API Sucker Rod Pumping Units Size 40 & 57

3.1.1.6 Rod Pump Operation

Fluid level detection

The pump capacity will often be greater than the well inflow capacity and the pump motor must be stopped at regular intervals when the fluid level is reduced to a specified, minimum safety level above the pump. This monitoring is performed with an "Echometer" attached to the wellhead, a tool which is consisted of a firing mechanism, a microphone and an amplifier recorder.

Typically, a gas actuated device generates an acoustic pulse at the wellhead which is then reflected back by subsurface items, with the main reflection to be from the fluid level at the bottom of the casing. The depth of the fluid level can now be found by multiplying this time by half the velocity of sound in the casing/ tubing annulus. The pump can be restarted when the casing fluid level has risen sufficiently due to inflow from the reservoir. The calibrated pump unit is getting on timer control as the “Echometer” is not being used continuously since well inflow performance normally shows a steady, predictable decline rate with time.

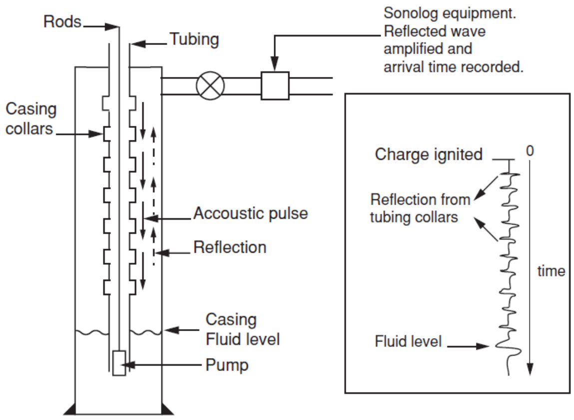


Figure3.1.1.6.a: "Echometer" or "Sonolog" fluid level survey

Gas Influx

The pump efficiency will be reduced when free gas is sucked into the pump. When the plunger moves upward some of the pump volume is lost due to expansion of the gas bubble in the pump chamber. A similar loss occurs due to the compression of the gas during the down stroke. A sufficiently large gas bubble results in a "Gas Lock" and the liquid pumping efficiency turns to zero. One thing that can be done is minimizing the volume between the travelling valve and standing valve at the bottom of the down stroke. In this way the gas is pushed out of the pump during each down stroke.
It is not always possible to place the pump below the perforations and many types of “gas anchors” have been tried in order to overcome the resulting difficulties. They all aim to separate the “free” gas from the liquid before it enters the pump barrel. The packer and crossover directs the multiphase flow above the pump and the gas flows into the tubing or casing annulus where it is vented.

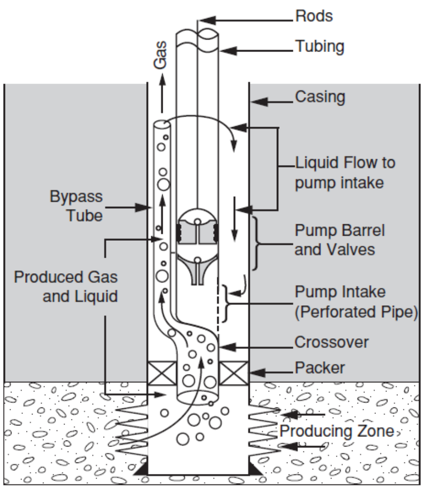


Figure 3.1.1.6.b: Packer type gas anchor transports the produced, multiphase fluid above the pump liquid intake

Centralizers

The sucker rods may require centralizers or protectors in deviated wells to reduce wear on the tubing and rods. This requirement becomes more extreme in crooked or highly deviated wells.

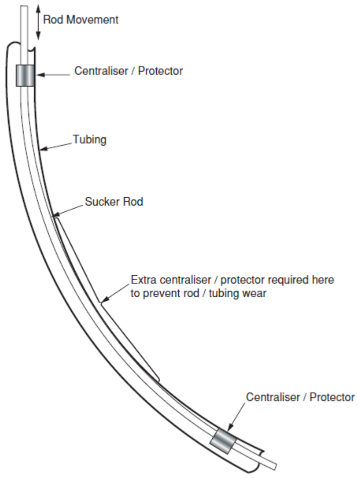


Figure3.1.1.6.c: Centralizing rod reduces friction and wear in a deviated well

Solids

Rod pumps have a very limited ability to lift sand due to the low fluid velocity in the production tubing and the wear on the pump valves, seats and plunger. Wax and inorganic scale deposition interfere with efficient rod pump operation. The removal of wax by hot oil/solvent circulation or injection of a scale inhibitor into the formation is possible and must be applied.

3.1.1.7 Principles of Pump Performance Analysis

The efficiency of sucker rod pumping units is usually analyzed using the information from pump dynagraph and polished rod dynamometer cards. A pump dynagraph is an instrument installed immediately above the plunger to record the plunger stroke and the loads carried by the plunger during the pump cycle.

- Card (a) shows an ideal case where instantaneous valve actions at the top and bottom of the stroke are indicated.
- Card (b) some free gas is drawn into the pump on the upstroke, so a period of gas compression can occur on the down-stroke before the TV opens.
- Card (c) shows gas expansion during the upstroke giving a rounding of the card just as the upstroke begins.
- Card (d) shows fluid pounding that occurs when the well is almost pumped off. This fluid pounding results in a rapid fall off in stress in the rod string and the sudden imposed shock to the system.
- Card (e) shows that the fluid pounding has progressed so that the mechanical shock causes oscillations in the system.
- Card (f) shows that the pump is operating at a very low volumetric efficiency and pump stroke is being lost in gas compression /expansion. Usually this gas-locked

condition is temporary and as liquid leaks past the plunger, the volume of liquid in the pump barrel increases until the TV opens and pumping recommences.

The use of the pump dynagraph involves pulling the rods and pump from the well bath to install the instrument and to recover the recording tube, thus it cannot be used in a well equipped with a tubing pump.

The dynagraph is more a research instrument than an operational device. So once there is knowledge from a dynagraph the surface dynamometer cards can be then interpreted.

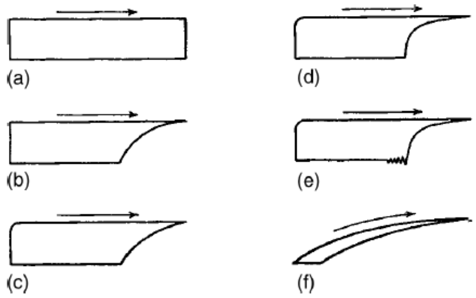


Figure 3.1.1.7.a: Pump dynagraph cards: (a) ideal card, (b) gas compression on down-stroke, (c) gas expansion on upstroke, (d) fluid pound, (e) vibration due to fluid pound, (f) gas lock (Nind, 1964)

The surface dynamometer is a device that records the motion and variations of the polished rod during the pumping cycle. The rod string is forced by the pumping unit to follow a regular time versus position pattern. However, the polished rod reacts with the loadings that are imposed by the well. The cards have three principal uses:

a. To obtain information that can be used to determine load, torque and horsepower changes required of the pump equipment
b. To improve pump operating conditions such as pump speed and stroke length
c. To check well conditions after installation of equipment to prevent or diagnose various operating problems

Correct interpretation of surface dynamometer card leads to estimate of various parameter values:

- Maximum and minimum PRLs can be read directly from the surface card (with the use of instrument calibration). These data then allow for the determination of the torque, counterbalance and horsepower requirements for the surface unit.
- Rod stretch and contraction is shown on the surface dynamometer card. This phenomenon is reflected in the surface unit dynamometer card
- Acceleration forces cause the ideal card to rotate clockwise. The PRL is higher at the bottom of the stroke and lower at the top of the stroke. In Fig. 3.1.1.7.b (b), Point A is at the bottom of the stroke.
- Rod vibration causes a serious complication in the interpretation of the surface card. This is result of the closing of the TV and the ‘‘pickup’’ of the fluid load by the rod string. This is the fluid pounding, a phenomenon which sets up damped oscillation in the rod string. These oscillations result in waves moving from one end of the rod string to the other. Because the polished rod moves slower near the top and bottom of the strokes, these stress (or load) fluctuations due to vibrations tend to show up more prominently at those locations on the cards. Figure 3.1.1.7.b (c) shows typical dynamometer card with vibrations of the rod string.

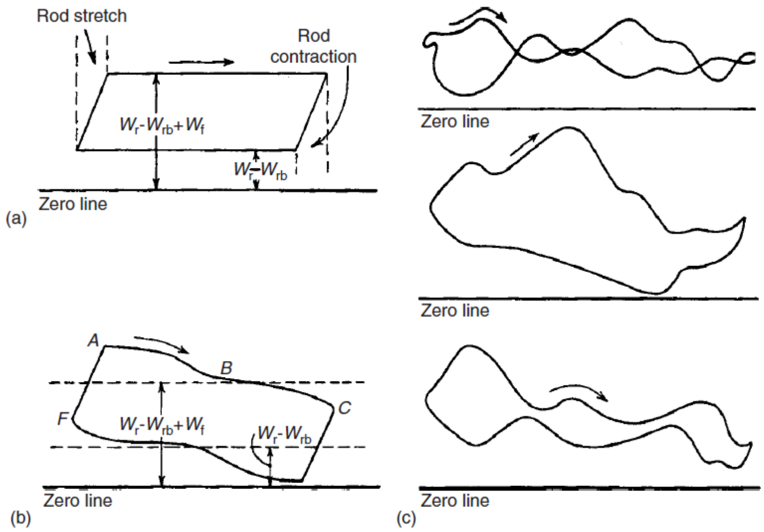


Figure 3.1.1.7.b: Surface dynamometer card: (a) ideal card (stretch and contraction), (b) ideal card (acceleration), (c) three typical cards (Nind, 1964)

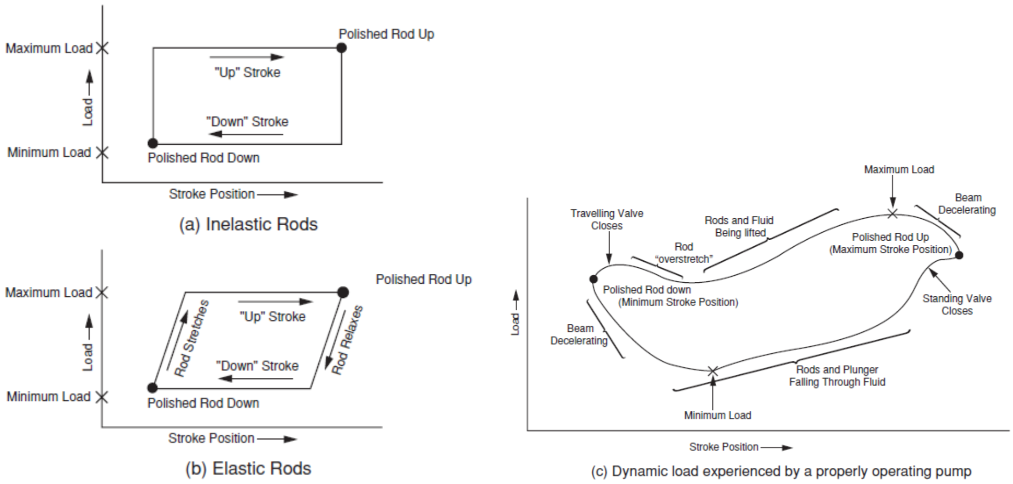


Figure 3.1.1.6.d: Dynamometer cards for: (a) inelastic rods (b) elastic rods (c) elastic rods with dynamic load of rods, fluid and surface pump unit pumping liquid only

- Figure 3.1.1.6 (a) records the variation in load for inelastic rods. The load is either high or low depending on whether the polished rod is moving up or down.
- Figure 3.1.1.6 (b) adds the elasticity of the rods: the increase to the maximum value in load is no longer instantaneous when the polished rod starts moving in a particular direction.
- Figure 3.1.1.6 (c) adds the further dimension of rod - fluid and surface pump unit dynamics. The times at which the various processes become controlling during the pump cycle are indicated.

The following effects alter the shape of the dynamometer trace in a distinctive manner allowing the source of the problem to be diagnosed and then rectified.

- Excessive rod or pump friction
- Restriction in the flow-path
- Vibrations
- Sticking plunger, leaking Travelling or Standing valves
- Presence of gas in the pump barrel, viscous emulsion formation etc.

3.1.1.8 Advantages and Disadvantages of Sucker Rod Pump

Advantages

- It is efficient, simple, and easy for field people to operate
- It can pump a well down to very low pressure to maximize oil production rate
- The system is easy to change to other wells with minimum cost
- Equipment from various supplies is fully interchangeable

Disadvantages

- Excessive friction in crooked/ deviated holes
- Solid-sensitive problems
- Low efficiency in gassy wells
- Limited depth due to rod capacity
- Bulky in offshore operations

3.1.2 Hydraulic Piston Pump

Hydraulic piston pumping systems can lift large volumes of liquid from great depth. They use as source of energy a high pressure power fluid pumped from the surface. They are applicable to multiple completions and offshore operations and crooked holes present minimal problems. Power fluid is consisted of oil or production water and supplied to the down hole equipment through a separate injection tubing.

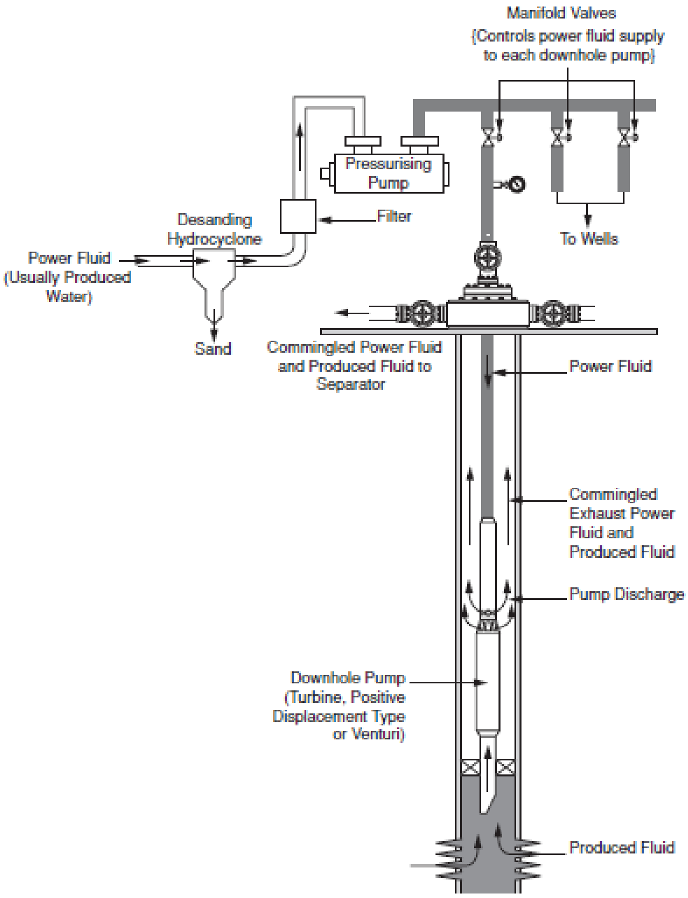


Figure 3.1.2: A hydraulic pump

3.1.2.1 Operation of a positive displacement hydraulic pump

A hydraulic piston pump (HPP) consists of an engine with a reciprocating piston driven by a power fluid connected by a short shaft to a piston in the pump end. HPPs are usually double-acting.

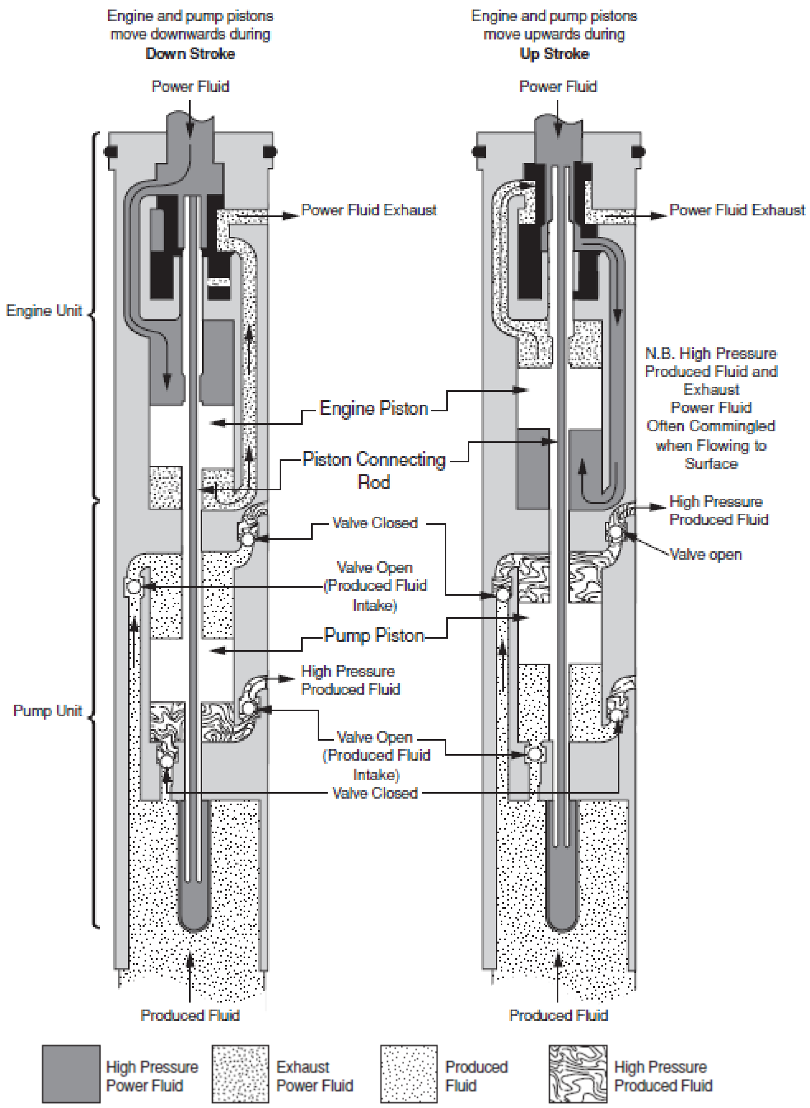


Figure3.1.2.1.a: Operation of a positive displacement hydraulic pump

The majority of installations commingle the exhaust fluid with the production fluid {an "Open" system Figure 3.1.2.1.b. (a)}. A "Closed" system may be installed if difficulties or high costs are encountered in preparing power fluid of the required quality from the production fluid. In this system the power fluid returns to the surface through a third separate tubing {Figure 3.1.2.1.b. (b)}. The completion design may also allow gas to be vented to surface via the casing/tubing annulus.

A typical power fluid supply pressure is between 1,500 and 4,000 psi and is provided by a pressurizing pump from the surface. This may be a triplex pump or a multi-stage, centrifugal pump. This pressure determines the pressure increase achievable by the down hole pump unit.

The power fluid must be "clean" to avoid erosion of the down hole pump components. It is often drawn from a settling tank where the larger solid particles are removed through a desanding hydrocyclone before having its pressure raised to the operating one.

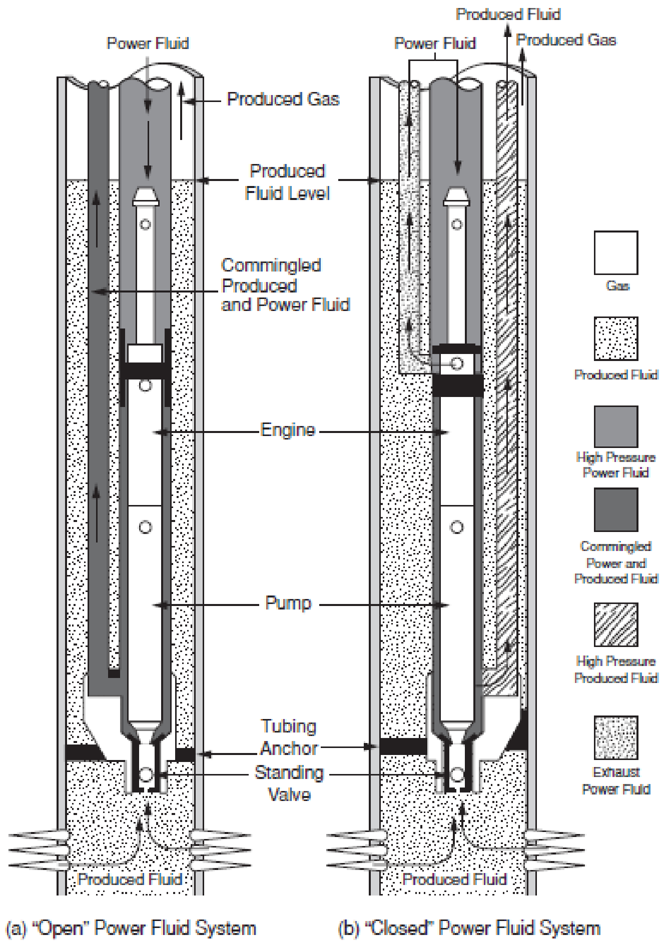


Figure 3.1.2.1.b: Types of hydraulic pump installation

3.1.2.2 Volumetric flow and proportionality factor

The pump rate is determined by the diameter and speed of the down hole pump. Engine pistons and pump are directly connected so the volumetric flow rates in the pump and engine are related through a simple equation (Cholet, 2000):

$$q_{pump} = q_{eng} \frac{A_{pump}}{A_{eng}} \tag{3.44}$$

where
q_{pump} = Flow rate of the produced fluid in the pump, bbl/day
q_{eng} = Flow rate of the power fluid, bbl/day
A_{pump} = Net cross-sectional area of pump piston, in^2
A_{eng} = Net cross-sectional area of engine piston, in^2

Equation (3.44) implies that liquid production rate is proportional to the power fluid injection rate. The proportionality factor $A_{pump} = A_{eng}$ is called the ''P/E ratio''. By adjusting the power fluid injection rate, the liquid production rate can be proportionally changed. Although the P/E ratio magnifies production rate, a larger P/E ratio means higher injection pressure of the power fluid.

3.1.2.3 Pressure relation

The following pressure relation can be derived from force balance in the HPP:

$p_{eng,i} - p_{eng,d} = (p_{pump,d} - p_{pump,i})\left({}^{P}/_{E}\right) + F_{pump}$ (3.45)

where

$p_{eng,i}$ = Pressure at engine inlet, psia
$p_{eng,d}$ = Engine discharge pressure, psia
$p_{pump,d}$ = Pump discharge pressure, psia
$p_{pump,i}$ = Pump intake pressure, psia
F_{pump} = Pump friction-induced pressure loss, psia

Equation (3.45) is also valid for open power fluid system where $p_{eng,d} = p_{pump,d}$

The pump friction-induced pressure loss F_{pump} depends on pump type, pumping speed, and power fluid viscosity. Its value can be estimated with the following empirical equation:

$$F_{pump} = 50\gamma_L(0.99 + 0.01v_{pf})(7.1e^{Bq_{total}})^{N/N_{max}} \quad (3.46)$$

where

γ_L = specific gravity of production liquid, 1.0 for H_2O
v_{pf} = viscosity of power fluid, centistokes
q_{total} = total liquid flow rate, bbl/day
N = pump speed, spm
N_{max} = maximum pump speed, spm
B = 0.000514 for $2\,{}^{3}/_{8}$ -in. tubing
0.000278 for $2\,{}^{7}/_{8}$ -in. tubing
0.000167 for $3\,{}^{1}/_{2}$ -in. tubing
0.000078 for $4\,{}^{1}/_{2}$ -in. tubing

The pump intake pressure $p_{pump,i}$ can be determined on the basis of well IPR and desired liquid production rate q_{Ld}. If the IPR follows Vogel's model, then for an HPP installed close to bottom hole, $p_{pump,i}$ can be estimated using

$$p_{pump,i} = 0.125\bar{p}\left[\sqrt{81 - 80\left({}^{q_{Ld}}/_{q_{max}}\right)} - 1\right] - G_b(D - D_p)$$

(3.47)

where

G_b = pressure gradient below the pump, psi/ft
D = reservoir depth, ft
D_p = pump setting depth, ft.

The pump discharge pressure $p_{pump,d}$ can be calculated based on wellhead pressure and production tubing performance. The engine discharge pressure $p_{eng,d}$ can be calculated based on the flow performance of the power fluid returning tubing. With all these parameter values known, the engine inlet pressure $p_{eng,i}$ can be calculated by Eq. (3.47). Then the surface operating pressure can be estimated by:

$$p_s = p_{eng,i} - p_h + p_f$$

where
p_s = surface operating pressure, psia
p_h = hydrostatic pressure of the power fluid at pump depth, psia
p_f = frictional pressure loss in the power fluid injection tubing, psi

The required input power can be estimated from the following equation:

$$HP = 1.7x10^{-5} q_{eng} p_s$$

Selection of HPP is based on the net lift defined by

$$L_{\boldsymbol{N}} = D_p - \frac{p_{pump,i}}{G_b} \tag{3.48}$$

and empirical value of P/E defined by

$$P/_{E} = \frac{10{,}000}{L_N} \tag{3.49}$$

3.1.2.4 Procedure for selecting an HPP

1. Starting from well IPR, the desirable liquid production rate q_{Ld} is determined. The pump intake pressure can then be calculated with Eq. (3.45).
2. Net lift is being calculated with Eq. (3.48) and P/E ratio with Eq. (3.49).
3. Calculation of flow rate at pump suction point by $q_{Ls} = B_o q_{Ld}$, where Bo is formation volume factor of oil, and then estimation of pump efficiencyE_p are possible.
4. The selection of pump rate ratio N/N_{max} must be between 0.2 and 0.8.The calculation of the design flow rate of pump is done by

$$q_{pd} = \frac{q_{Ls}}{E_p(N/N_{max})}$$

5. Based on q_{pd} and P/E values, a pump is selected from the manufacturer's literature and get rated displacement values q_{pump},q_{eng}, and N_{max}. If not provided, calculated flow rates per stroke are given by

$$q'_{pump} = \frac{q_{pump}}{N_{max}}$$

and

$$q'_{eng} = \frac{q_{eng}}{N_{max}}$$

6. Pump speed calculations by

$$N = (\frac{N}{N_{max}})N_{max}$$

7. Power fluid rate calculations by

$$q_{pf} = (\frac{N}{N_{max}})\frac{q_{eng}}{E_{eng}}$$

8. The return production flow rate is determined by

$$q_{total} = q_{pf} + q_{Ls} \quad \text{for open power fluid system or}$$
$$q_{total} = q_{Ls} \quad \text{for closed power fluid system}$$

9. Pump and engine discharge pressure $p_{pump,d}$ and $p_{eng,d}$ calculations are based on tubing performance.
10. Pump friction-induced pressure loss calculations by using Eq. (3.46).
11. Required engine pressure calculations by using Eq. (3.45).
12. The calculation of pressure change Δp_{inj} from surface to engine depth in the power fluid injection tubing based on single-phase flow has two components:

$$\Delta p_{inj} = p_{potential} + p_{friction}$$

13. Required surface operating pressure is calculated by

$$p_{so} = p_{eng,i} - \Delta p_{inj}$$

14. Required surface operating horsepower is calculated by

$$HP_{so} = 1.7x10^{-5}\frac{q_{pf}p_{so}}{E_s}$$

where Es is the efficiency of surface pump

3.1.2.5 Advantages and Disadvantages of a Hydraulic Pumps

Advantages

- Suitable for crooked and deviated wells.
- Reciprocating and turbine pumps can work at great depths (up to 17,000 ft).
- Very flexible speed control by the (surface) supply of power fluid. Turndown to < 20% of design maximum speed can be achieved.
- Turbine pumps can be manufactured from erosion resistant materials which gives an increased lifetime with "reasonable" solids production. They have proven to be very reliable for operation in remote locations where access to the wellhead is limited. It was also found possible to use the same water source (and many common flow lines) as used for reservoir pressure maintenance.
- The power source is remote from the wellhead giving a low wellhead profile, attractive for offshore and urban locations.
- The power fluid can carry corrosion or other inhibitors down hole, providing continuous inhibition when the well is producing.
- The pump unit can be designed as a "free" pump; the pump unit having the capability of being pumped through the power fluid tubing from the surface to its down hole location. It can then be recovered by reversing the flow direction. The ability to recover the pump without the need to move a rig/work over hoist to the well site is attractive for offshore platforms as well as remote and urban locations.

Disadvantages

- Pumps with moving parts have a short run life when supplied with poor quality (solids containing) power fluid
- Positive displacement and centrifugal pumps can achieve very low flowing bottom hole pressures in the absence of a gas effect
- A similar volume of power fluid and produced fluid is required, increasing the size of the production separators
- Power oil systems are fire hazards and costly
- Power water treatment problems
- High solids production is troublesome

3.1.3 Plunger Lift

Plunger lift systems are applicable to high GOR wells. The purpose of plunger lift is like that of other artificial lift methods: to remove liquids from the wellbore so that the well can be produced at the lowest bottom-hole pressures.

Traditionally, plunger lift was used on oil wells but recently they become more common on gas wells for de-watering purposes. Hi-pressure gas wells produce gas carrying liquid water and condensate in the form of mist. As the gas flow velocity in the well drops as a result of the reservoir pressure depletion, the carrying capacity of the gas decreases. When the gas velocity drops to a critical level, liquid begins to accumulate in the well and the well flow can undergo annular flow regime followed by a slug flow regime.

The liquid loading increases bottom-hole pressure that reduces gas production rate and furthermore low gas production rate will cause gas velocity to drop. Eventually the well will undergo bubbly flow regime and cease producing.

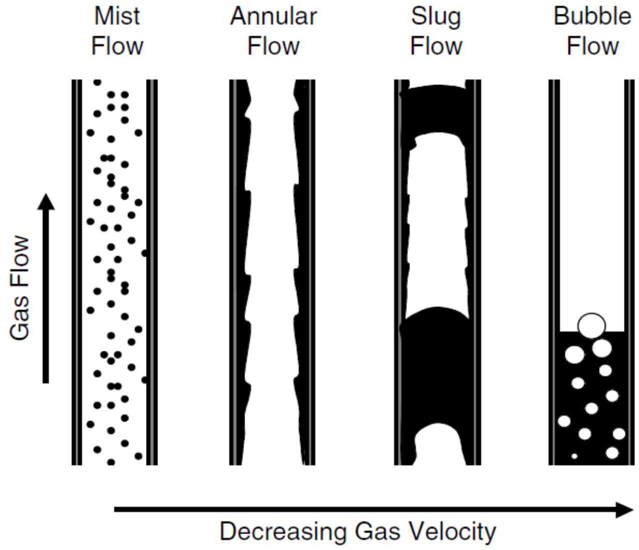


Figure 3.1.2: Four flow regimes commonly encountered in gas wells

Recognizing the liquid loading problem is not an obvious and easy task. A thorough diagnostic analysis of well data as liquid slugs at the surface of well, increasing difference between the tubing and casing pressures with time, sharp changes in gradient on a flowing pressure survey and sharp drops in a production decline curve needs to be performed and prediction with analytical methods must carried out.

Turner et al. (1969) were the pioneer investigator who analyzed and predicted the minimum gas flow rate capable of removing liquids from the gas production wells.
Starting from the Turner et al. entrained drop model, Guo and Ghalambor (2005) determined the minimum kinetic energy of gas that is required to lift liquids. A four-phase (gas, oil, water, and solid particles) mist flow model was developed. The application of the minimum kinetic energy criterion to the four-phase flow model resulted in a closed-form analytical equation for predicting the minimum gas flow rate.

3.1.3.1 Working Principle

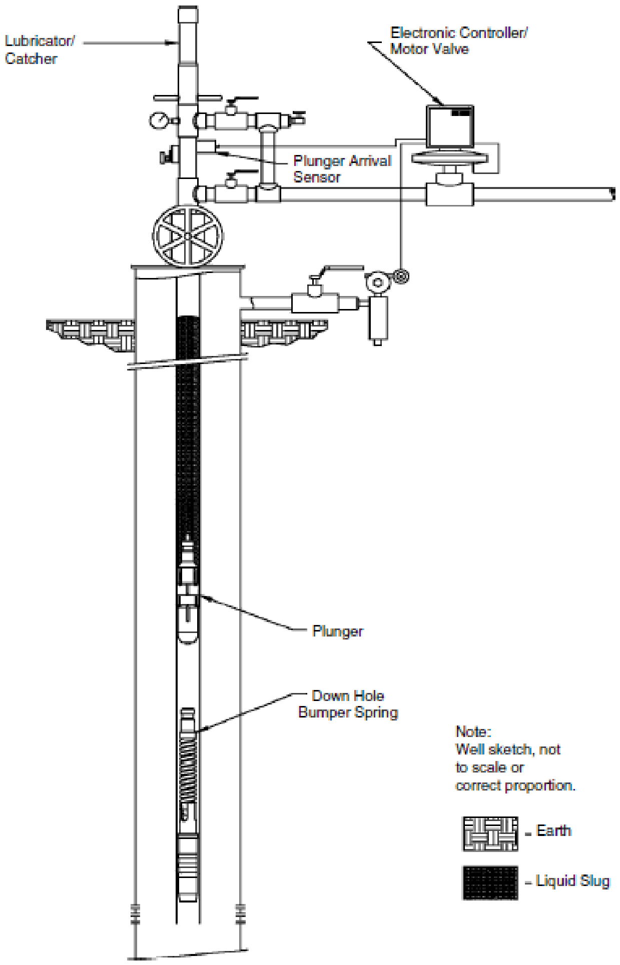


Figure 3.1.3.1: A sketch of a plunger lift system (courtesy Ferguson Beauregard)

Plunger lift uses a free piston that travels up and down in the well's tubing string. It minimizes liquid fallback and uses the well's energy more efficiently than in slug or bubble flow.
Whether in a gas well, oil well or gas lift well, the mechanics of a plunger lift system are the same. The plunger, a length of steel, is dropped down the tubing to the bottom of the well and allowed to travel back to the surface. It provides a piston-like interface between liquids and gas in the wellbore and prevents liquid fallback. By providing a ''seal'' between the liquid and gas, a well's own energy can be used to efficiently lift liquids out of the wellbore.
However, in a well without a plunger, gas velocity must be high to remove liquids. With a plunger, gas velocity can be very low. Unloading relies much more on the well's ability to

store enough gas pressure to lift the plunger and a liquid slug to surface and less on critical flow rates.

Plunger operation consists of shut-in and flow periods. The flow period is further divided into an unloading period and flow after plunger arrival. The lengths of these periods will vary up on the application, producing capability of the well and pressures.

Pumping Cycle-Shut-in Period

A cycle starts with the shut-in period that allows the plunger to drop from the surface to the bottom of the well. At the same time, the well builds gas pressure stored either in the casing, in the fracture or in the near wellbore region of the reservoir. The well must be shut in long enough to build reservoir pressure that will provide energy to lift both the plunger and the liquid slug to the surface against line pressure and friction. When this time and pressure have been reached, the flow period is started and unloading begins.

Pumping Cycle -Flow Period

In the initial stages of the flow period the plunger and liquid slug are begin traveling to the surface. Gas above the plunger quickly flows from the tubing into the flow line, and the plunger and liquid slug follow up the hole. The plunger arrives at the surface and unloads the liquid. Initially high rates are prevailing, three to four times the average daily rate, while the stored pressure is blown down. The well can now produce free of liquids, while the plunger remains at the surface, held by the well’s pressure and flow. As rates drop, velocities eventually drop below the critical rate and liquids begin to accumulate in the tubing. The well is shut in and the plunger falls back to the bottom to repeat the cycle.

Pressure Behavior

At the end of the shut-in period, the well has built pressure. The casing pressure is at its maximum and the tubing pressure is lower than the casing pressure. The difference is equivalent to the hydrostatic pressure of the liquid in the tubing. When the well is opened, the tubing pressure quickly drops down to line pressure, while the casing pressure slowly decreases until the plunger reaches the surface.

When the plunger is near the surface, the liquid on top of the plunger may surge through the system, causing spikes in line pressure and flow rate. This continues until the plunger reaches the surface. After the plunger surfaces, a large increase in flow rate will produce higher tubing pressures and an increase in flow line pressure. Tubing pressure will then drop very close to line pressure. Casing pressure will reach its minimum either on plunger arrival or after, as the casing blows down and the well produces with minimal liquids in the tubing. If the well stays above the critical unloading rate, the casing pressure will remain fairly constant or may decrease further. As the gas rate drops, liquids become held up in the tubing and casing pressure will increase.

Upon shut in, the casing pressure builds more rapidly and depends on the inflow performance and reservoir pressure of the well. The tubing pressure will increase quickly from line pressure, as the flowing gas friction ceases. It will eventually track casing pressure (less the liquid slug). Casing pressure will continue to increase to maximum pressure until the well is opened again. As with most wells, maximum plunger lift production occurs when the well produces against the lowest possible bottom-hole pressure. On plunger lift, the lowest average bottom-hole pressures are almost always obtained by shutting the well in the minimum amount of time.

Practical experience and plunger lift models demonstrate that lifting large liquid slugs requires higher average bottom-hole pressure. Lengthy shut-in periods also increase average bottom-hole pressure. So the goal of plunger lift should be to shut the well in the minimum amount of time and produce only enough liquids that can be lifted at this minimum buildup pressure.

Shut-in time

The absolute minimum amount of time for shut-in is the time it takes the plunger to reach the bottom. The well must be shut-in in this length of time regardless of what other operating conditions exist. Plungers typically fall between 200 and 1,000 ft/min in dry gas and 20 and 250 ft/min in liquids and total fall time varies and is affected by plunger type, amount of liquids in the tubing, the condition of the tubing and the deviation of the tubing or wellbore.

Flow period

The flow period during and after plunger arrival is used to control liquid loads. In general, a short flow period brings in a small liquid load and a long period a larger liquid load. By controlling this flow time, the liquid load can be controlled and the well can be flowed until the desired liquid load has entered the tubing. A well with a high GLR may be capable of long flow periods without requiring more than minimum shut-in times. In this case, the plunger could operate as few as 1 or 2 cycles/ day. Conversely, a well with a low GLR may never be able to flow after plunger arrival and may require 25 cycles/day or more.

In practice, if the well is shutting in for only the minimum amount of time, it can be flowed as long as possible to maintain target plunger rise velocities. If the well is shutting in longer than the minimum shut-in time, there should be little or no flow after the plunger arrives at the surface.

3.1.3.2 Design Guideline

Plunger lift systems can be evaluated using rules of thumb in conjunction with historic well production or with a mathematical plunger model. Because plunger lift installations are typically inexpensive, easy to install, and easy to test, most evaluations are performed by rules of thumb.

Estimate of Production Rates with Plunger Lift

The simplest and most accurate method of determining production increases from plunger lift is from decline curve analysis. Gas and oil reservoirs typically have predictable declines, exponential, harmonic or hyperbolic. Initial production rates are usually high enough to produce the well above critical rates and establish a decline curve. When liquid loading occurs, a marked decrease and deviation from normal decline can be seen. By unloading the well with plunger lift, a normal decline can be reestablished.

Production increases from plunger lift will be somewhere between the rates of the well when it started loading and the rate of an extended decline curve to the present time. Ideally, decline curves would be used in concert with critical velocity curves to predetermine when plunger lift should be installed. In this manner, plunger lift will maintain production on a steady decline and never allow the well to begin loading. Another method to estimate production is to build an inflow performance curve based on the backpressure equation.
This is especially helpful if the well has an open annulus and casing pressure is known. The casing pressure gives a good approximation of bottom-hole pressure. The IPR curve can be built based on the estimated reservoir pressure, casing pressure and current flow rate. Because the job of plunger lift is to lower the bottom-hole pressure by removing liquids, the bottom-hole pressure can be estimated with no liquids. This new pressure can be used to estimate a production rate with lower bottom-hole pressures.

GLR and Buildup Pressure Requirements

There are two minimum requirements for plunger lift operation: minimum GLR and buildup pressure. For the plunger lift to operate there must be available gas to provide the lifting force, in sufficient quantity per barrel of liquid for a given well depth.

Rules of Thumb

As a rule of thumb, the minimum GLR requirement is considered to be about 400 scf/bbl/1,000 ft of well depth, that is

$$GLR_{min} = 400\frac{D}{1{,}000}$$

(3.50)
where
GLR_{min} = minimum required GLR for plunger lift, scf/bbl
D = depth to plunger, ft.

Equation (3.50) is based on the energy stored in a compressed volume of 400 scf of gas expanding under the hydrostatic head of a barrel of liquid. The drawback is that no consideration is given to line pressures. Excessively high line pressures, relative to buildup pressure may increase the requirement. The rule of thumb also assumes that the gas expansion can be applied from a large open annulus without restriction. Slim-hole wells and wells with packers that require gas to travel through the reservoir or through small perforations in the tubing will cause a greater restriction and energy loss. This increases the minimum requirements to as much as 800–1,200 scf/bbl/1,000 ft.
Well buildup pressure is the second requirement for plunger operation. This buildup pressure is the bottom hole pressure just before the plunger begins its ascent (equivalent to surface casing pressure in a well with an open annulus). In practice, the minimum shut-in pressure requirement for plunger lift is equivalent to 1½ times maximum sales line pressure. The actual requirement may be higher. The rule works well in intermediate-depth wells (2,000–8,000 ft) with slug sizes of 0.1–0.5 barrels/cycle. It breaks down for higher liquid volumes, deeper wells (due to increasing friction), and excessive pressure restrictions at the surface or in the wellbore. An improved rule for minimum pressure is that a well can lift a slug of liquid equal to about 50–60% of the difference between shut-in casing pressure and maximum sales line pressure. This rule gives:

$$p_c = p_{Lmax} + \frac{p_{sh}}{f_{sl}} \quad (3.51)$$

where
p_c = required casing pressure, psia
p_{Lmax} = maximum line pressure, psia
p_{sh} = slug hydrostatic pressure, psia
f_{sl} = slug factor, 0.5–0.6

This rule takes liquid production into account and can be used for wells with higher liquid production that require more than 1–2 barrels/cycle. It is considered as a conservative estimate of minimum pressure requirements. To use Eq. (3.51), the total liquid production on plunger lift and number of cycles possible per day should be estimated.
Then the amount of liquid that can be lifted per cycle should be determined. That volume of liquid per cycle is converted into the slug hydrostatic pressure using the well tubing size.

Finally, the equation is used to estimate required casing pressure to operate the system.
It should be noted that a well that does not meet minimum GLR and pressure requirements could still be plunger lifted with the addition of an external gas source. Design at this point becomes more a matter of the economics of providing the added gas to the well at desired pressures.

3.1.3.3 Advantages and Disadvantages of Plunger Lift

Advantages

- Retrievable without pulling tubing
- Very inexpensive installation
- Automatically keeps tubing clean of paraffin and scale
- Applicable for high GOR wells
- Can be used with intermittent gas lift
- Can be used to unload liquid from gas wells

Disadvantages

- May not take well to depletion; therefore, eventually requires another lift method
- Good for low-rate, normally less than 200 B/D wells only
- Requires more engineering supervision to adjust properly
- Danger exists in plunger reaching too high velocity and causing surface damage
- Communication between tubing and casing required for good operation unless used in conjunction with gas lift

3.1.4 Progressing Cavity Pump

Progressing Cavity Pumps are becoming increasingly popular for the production of viscous crude oils. Many applications are at shallow wells but it is also possible to design PCP's with production rates between 5 to 5,000 bfd producing from depths >8,000ft TVD and with a maximum pressure differential across the pump of greater than 5,000 psi.
A typical completion is illustrated in Figure 3.1.4 where a prime mover is shown rotating a sucker rod string and driving the PCP.

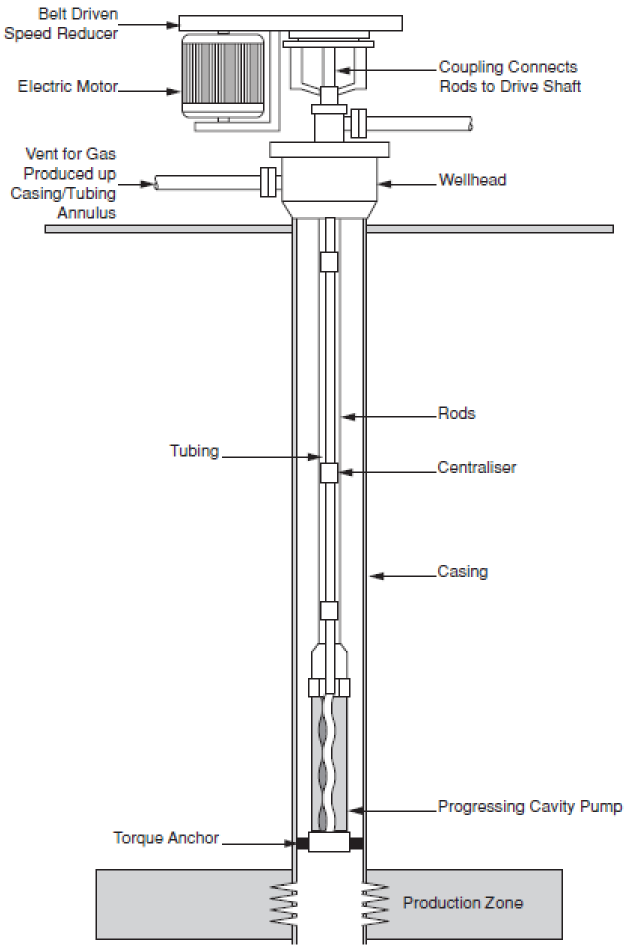


Figure 3.1.4: A well completed with artificial lift using a progressing cavity pump

3.1.4.1 Progressing Cavity Pump Principle

The progressive cavity pump (PCP) is a positive displacement pump, using an eccentrically rotating single-helical rotor, turning inside a stator. The rotor is rotated inside an elastomeric pump body or stator, which has been molded in the form of a double helix with a pitch of the same diameter and exactly twice the length of the pitch given to the rotor. Figure 3.1.4.a illustrates the main components of a PCP. When assembled, the center line of the rotor and the stator are slightly offset, creating a series of fluid filled cavities along the length of the pump. Figure 3.1.4.b is a perspective view of the stator, which helps explain how the interference fit between the rotor and stator creates two chains of spiral (fluid filled) cavities.

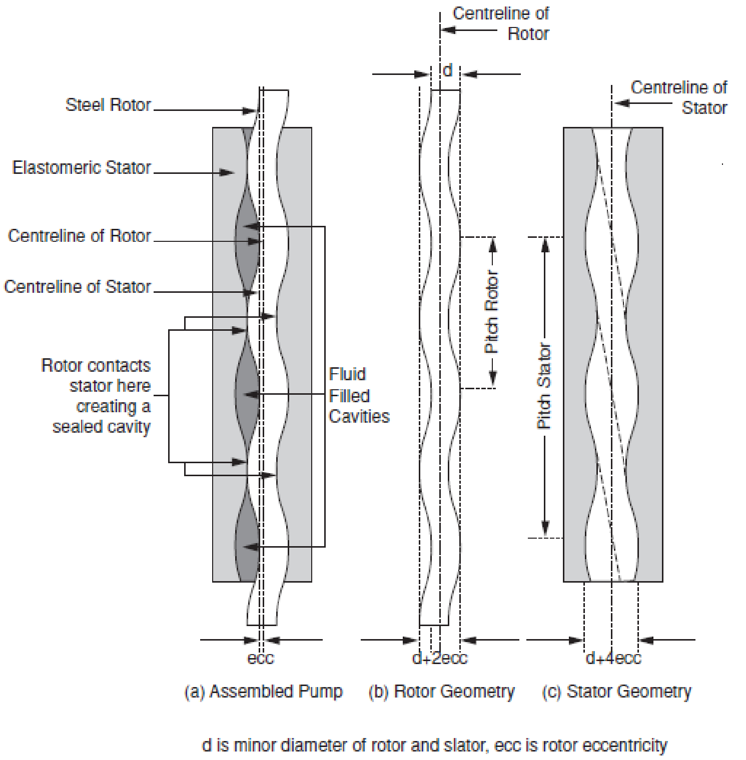


Figure3.1.4.1.a: Cross section progressing cavity pump and its components

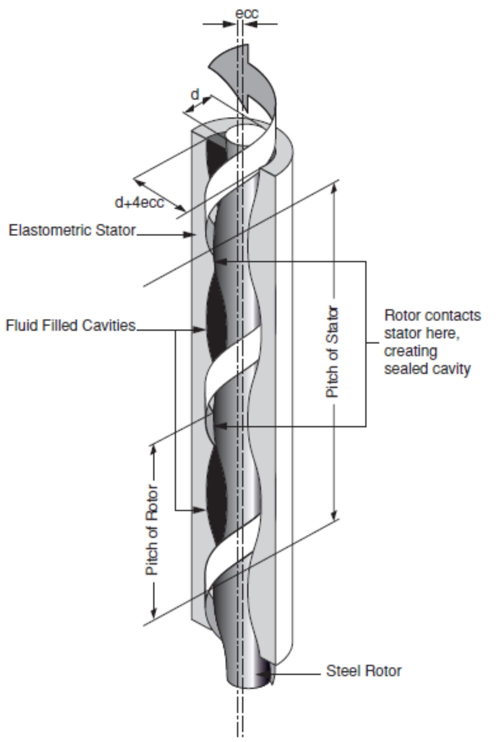


Figure 3.1.4.1.b: Perspective view of operating Progressing Cavity Pump

The rotation of rotor within the stator operates as a positive displacement pump. This causes the trapped fluid in the sealed cavities, to progress along the length of the pump from the suction to the pump discharge. The size and shape of the cavities do not change during this progression. Figure 3.1.4.1.c shows how, as one cavity diminishes, the next one increases at exactly the same rate and gives a constant, non-pulsating flow.

The number of “seal-lines” formed along the pump body by the rotor and stator will determine the pressure increase that can be achieved by the pump.
The head generated by the pump will increase as the rotational speed of the stator is increased. When the head increases above the required per stage level, the fluid will “slip” backwards if a greater pressure increase is demanded from the pump and the flow rate increase will no longer be proportional to the increase in the stator's rotational speed.
This fluid "slip" can be avoided by increasing the number of pump stages. Wear of either the stator of rotor will decrease this value since the maximum pressure increase depends on this interference fit. The construction of the stator body from an elastomer makes this pump design relatively tolerant to produced solids - particularly since they are often used to pump viscous oils which provide a lubrication film to protect the rotor and stator from wear.

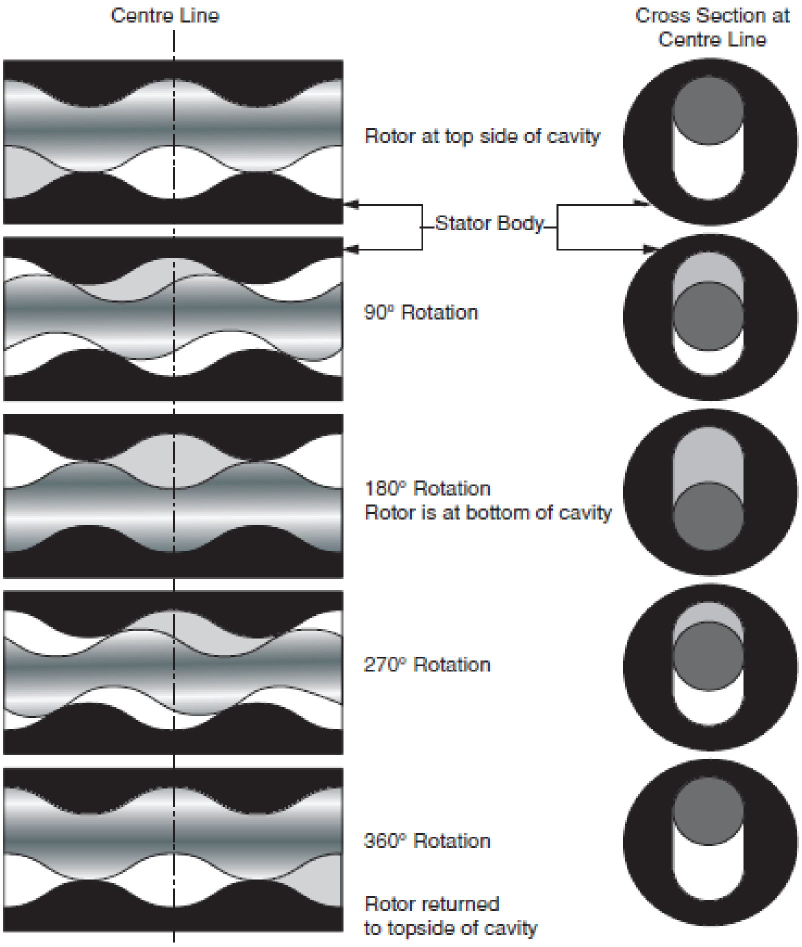


Figure 3.1.4.1.c: Operating principle of Progressing Cavity Pump

3.1.4.2 Down-Hole PCP Characteristics

Proper selection of a PCP requires knowledge of PCP geometry, displacement, head and torque requirements. Figure 3.1.4.2 illustrates rotor and stator geometry of PCP
where
D = rotor diameter, in.
E = stator eccentricity, in.
P_r = pitch length of rotor, ft
P_s = pitch length of stator, ft.

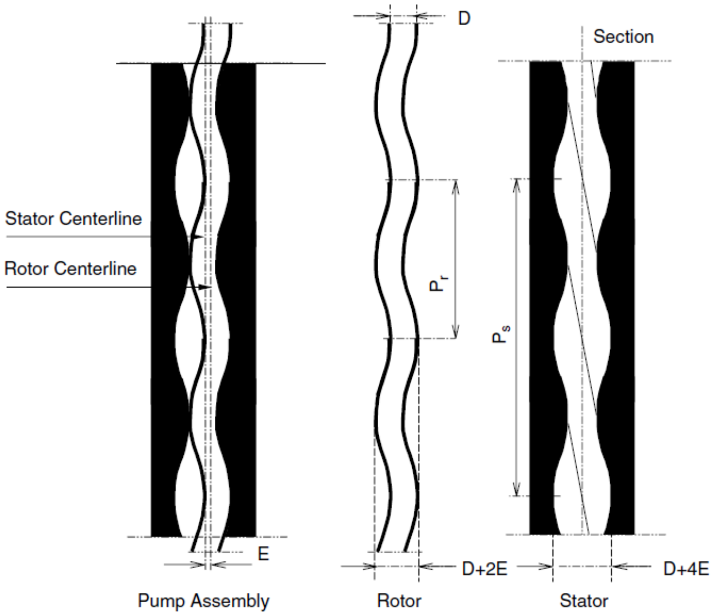


Figure 3.1.4.2: Rotor and stator geometry of PCP (Cholet, 2000)

Two numbers define the geometry of the PCP: the number of lobes of rotor and the number of lobes of the stator. A pump with a single helical rotor and double helical stator is described as a ‘‘1-2 pump’’ where $P_s = 2P_r$. For a multi-lobe pump:

$$P_s = \frac{L_r + 1}{L_r} P_r$$

where L_r is the number of rotor lobes. The ratio ${}^{P_r}/_{P_s}$ is called the ‘‘kinematics ratio’’.

Pump displacement is defined by the fluid volume produced in one revolution of the rotor:

$V_o = 0.028ED\ P_s$

The flow rate achieved by a liquid filled (no gas) PCP pump is directly proportional to the speed of rotation of the rotor (N). Pump flow rate is expressed as:

$$Q_c = 7.12EDP_sN - Q_s$$

where
V_o = pump displacement, ft^3
Q_c = pump flow rate, bbl/day
N = rotary speed, rpm
Q_s = leak rate, bbl/day
The volumetric efficiency is typically 70-80%. The presence of gas will reduce this efficiency, so a gas anchor is being frequently included in PCP completions.

The PCP head rating is defined by:

$$\Delta P = (2n_p - 1)\delta p \qquad (3.52)$$

where
ΔP = pump head rating, psi
n_p = number of pitches of stator
δp = head rating developed into an elementary cavity, psi

PCP mechanical resistant torque is expressed as:

$$T_m = \frac{144 V_o \Delta P}{e_p} \tag{3.53}$$

where
T_m =mechanical resistant torque, lb_f-ft
e_p = efficiency

The load on thrust bearing through the drive string is expressed as

$$F_b = \frac{\pi}{4}(2E + D)^2 \Delta P \tag{3.54}$$

where F_b = axial load, lb_f

3.1.4.3 Selection of various components

The following procedure can be used in the selection of a PCP:

1. Starting from well IPR, a desirable liquid flow rate q_{Lp} is selected at pump depth and the corresponding pump intake pressure below the pump p_{pi}
2. Based on manufacturer's literature, a PCP is selected that can deliver liquid rate Q_{Lp}, where $Q_{Lp}, > q_{Lp}$. Obtain the value of head rating for an elementary cavity δp
3. The required pump discharge pressure p_{pd} is determined based on wellhead pressure, tubing size, flow rate Q_{Lp}, and fluid properties
4. Required pump head is being calculating by $\Delta P = p_{pd} - p_{pi}$
5. The required number of pitches n_p is being calculated by using Eq.(3.52)
6. Mechanical resistant torque is being calculated by using Eq.(3.53)
7. The load on thrust bearing is being calculated by using Eq.(3.54)

Selection of Drive String

Sucker rod strings used in beam pumping are also used in the PCP systems as drive strings. The string diameter should be properly chosen so that the tensile stress in the string times the rod cross-sectional area does not exceed the maximum allowable strength of the string.

The following procedure can be used in selecting a drive string:

1. The weight of the selected rod string W_r in the effluent fluid (liquid level in annulus should be considered to adjust the effect of buoyancy) must be calculated
2. The thrust generated by the head rating of the pump F_b must be calculated with Eq. (3.54)
3. Mechanical resistant torque T_m must be calculated with Eq. (3.53)
4. Calculate the torque generated by the viscosity of the effluent in the tubing by:

$$T_v = 2.4 \, x \, 10^{-6} \mu_f L N \frac{d^3}{(D-d)} \frac{1}{ln \frac{\mu_s}{\mu_f}} (\frac{\mu_s}{\mu_f} - 1)$$

where
T_v = viscosity-resistant torque, lbf -ft
μ_f = viscosity of the effluent at the inlet temperature, cp
μ_s = viscosity of the effluent at the surface temperature, cp
L = depth of tubing, ft
d = drive string diameter, in

5. Total axial load to the drive string must be calculated by: $F = F_b + W_r$
6. Total torque must be calculated by: $T = T_m + T_v$
7. The axial stress in the string must be calculated by:

$$\sigma_\tau = \frac{4}{\pi d^3}\sqrt{F^2 d^2 + 64T^2 x\ 144}$$

where the tensile stress σ_τ is in pound per square inch. This stress value should be compared with the strength of the rod with a safety factor.

Selection of Surface Driver

The prime mover for PCP can be an electrical motor, hydraulic drive or internal-combustion engine. The minimum required power from the driver depends on the total resistant torque requirement from the PCP, that is,

$$P_h = 1.92\ x\ 10^{-4} TN$$

where the hydraulic power P_h is in hp. Driver efficiency and a safety factor should be used in driver selection from manufacturer's literature.

3.1.4.4 Progressing Cavity Pump Operational Issues

PCP's have been powered by an electric motor and gearbox mounted above the wellhead that rotates a string of sucker rods connected to the PCP pump in most cases. The rods are rotated rather than reciprocated as used to power Rod Pumps. The sucker rod string is susceptible to failure, especially in crooked or deviated wells or when formation sand is being produced along with the produced fluid. There is also a tendency to a higher frequency of tubing failures since the rods are rotating against the inner wall of the tubing.

Surface driven PCPs are often installed in deviated wells and the pump is being landed in well sections with a deviation in excess of 70 degrees in some cases. Note that the dogleg severity should also be controlled; a value of < 5º/100 ft is being recommended for normal PCP performance.

Rod/tubing frictional contact, even when reduced by centralising the sucker rod string, leads to a high starting torque as well as wasting power when the pump is operating. The use of rod centralizers to reduce the side loading of the rods against the tubing is a necessity in deviated wells. A small diameter, production tubing has a reduced clearance between the rod couplings and tubing's inner wall. An increased number of non-rotating couplings is required to manage the contact forces between the rod string and tubing. An alternative is to reduce this side loading further by running a continuous rod without couplings.

The required pressure boost to be generated by the pump needs to be carefully calculated. For example, the pressure drop across the centralizers and the extra friction and hydrostatic head associated with denser crudes, needs to be considered along with the necessary reservoir drawdown to produce at the planned production rate.

Failure of the pump, the sucker rod string or the tubing requires that all three have to be pulled to the surface and then rerun to return the well to production. This operation can normally be done by a light work over hoist, since the PCP pumped wells are normally shallow and are not capable of natural flow.

A long PCP pump life requires that a compatibility test with a representative oil sample with the elastomer used for construction of the stator has been conducted. Issues that need to be taken into account in specification of the elastomer include the Bottom Hole Temperature, aromatic compounds present in the oil, possible chemical treatments that may be performed in the well (injection of scale or corrosion inhibitor, acid etc.), bottom hole pressure relative to fluid bubble point, contaminants H_2S and CO_2.

The PCP pump should not be run in a "pumped-off" condition. The fluid level in the annulus should at all times be above the pump inlet. Down hole pressure gauges are the ultimate "pump-off" protection; though thousands of PCPs operate without such gauges. Monitoring the liquid level in the annular space using a sonic device may also be done. The PCP is a positive displacement pump; hence monitoring the surface pressure or motor current will also indicate if "pump-off" is occurring.

3.1.4.5 Advantages and Disadvantages of Progressing Cavity Pump

Advantages

- Simple design and rapid repair of pump unit by replacing rotor and stator as a complete unit
- High volumetric efficiency in the absence of gas
- Tolerance in production solids at reasonable levels
- High energy efficiency
- Emulsions are not forming due to low shear pumping action
- Capable of pumping viscous crude oil
- Used for lifting heavy oils at a variable flow rate
- They can be installed in deviated and horizontal wells

Disadvantages

- High staring torque
- Fluid compatibility problems with elastomers in direct contact with aromatic crude oil
- Gas dissolves in the elastomers at high bottom hole pressure
- Short operating life (2–5 years)
- High cost

3.2 Dynamic Displacement Pumps

3.2.1 Electric Submersible Pumps (ESPs)

Electric submersible pumps (ESPs) are easy to install and operate. They can operate over a wide range of depth and volume parameters reaching depths of 12,000 ft and volumetric flow rates up to 45,000 bbl/day.

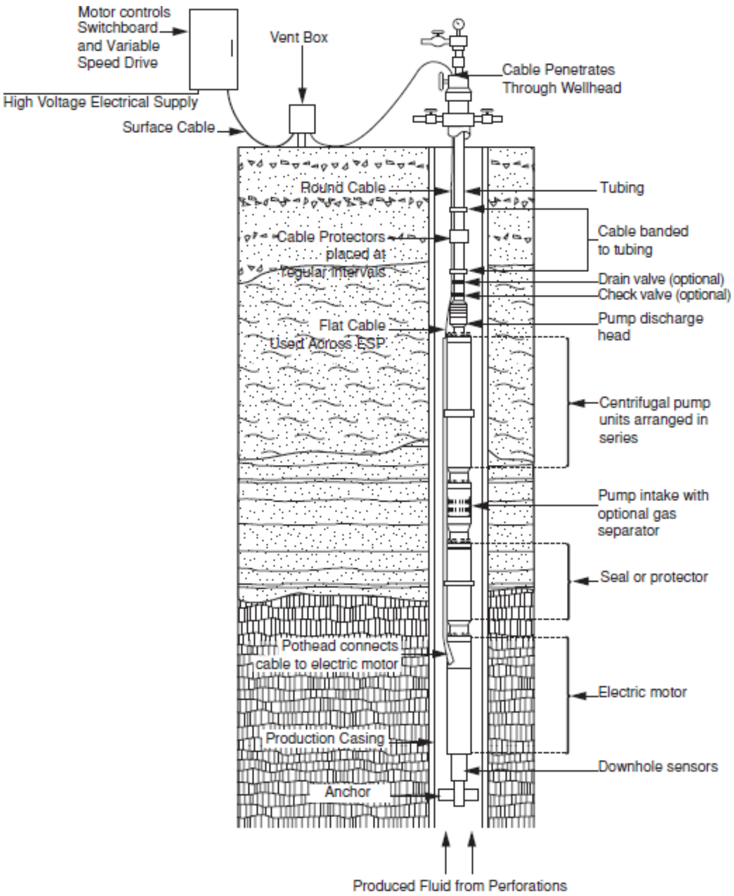


Figure 3.2.1.1: Well completed with an electric submersible centrifugal pump

3.2.1.1 Subsurface and surface components

a. **Surface components**
- Surface motor controller
- Surface electric cable
- Vent box

b. **Subsurface components**
- Down hole cable
- Pump
- Seal section
- Motor
- Optional components

3.2.1.2 ESP Principle

ESPs are pumps made of dynamic pump or centrifugal pump stages. Figure 3.2.1.2.a gives the internal schematic of a single-stage centrifugal pump and a cutaway of a multistage centrifugal pump. The electric motor connects directly to the centrifugal pump as the electric motor shaft connects directly to the pump shaft. Thus, the pump rotates at the same speed as the electric motor.

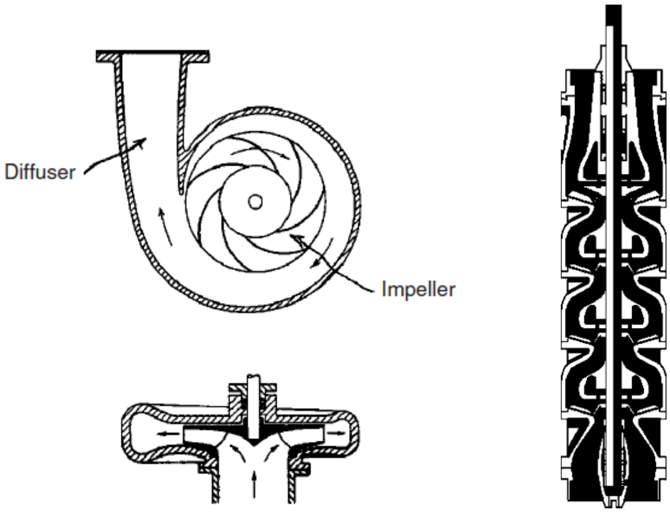


Figure3.2.1.2.a: An internal schematic of centrifugal pump and a sketch of a multistage centrifugal pump

In ESP operations, electric energy is transported to the down-hole electric motor via the electric cables. These electric cables are run on the side of the production tubing and provide the electrical energy needed to actuate the down-hole electric motor. The electric motor drives the pump and the pump imparts energy to the fluid in the form of hydraulic power, which lifts the fluid to the surface. The functions of the surface and subsurface components are summarized as follows:

Motor controller

A high voltage, three phase electricity supply is transformed to the “nameplate” voltage of the ESP. The supply frequency can be changed using a Variable Frequency Drive (VFD) which allows the speed of the electric motor to be altered. A VFD also allows the pump flow rate to be adjusted to the well inflow conditions (changes in reservoir pressure and skin) as these alter during the life of the well.

Vent Box

The vent box is a safety equipment that separates the surface cable from the down hole cable. This ensures that any gas which diffuses up the down hole cable will be dissolved in the rubber insulation and will not reach the electrical switchgear. Otherwise an explosion due to a spark generated when switching the pump on or off is possible.

Down hole Cable

The down hole cable has a “flat pack” shape in contrast with the round shape of the surface cable and penetrates the well through the wellhead. It is banded to the tubing at regular intervals and additional protection is supplied by cable protectors, which are installed at critical points to prevent damage while the completion is being run into the hole. The cable enters the electric motor housing at the Pothead carrying the electrical power supply for the motor and the measurement signal from the down hole sensor package installed underneath the motor.

The Pump

The pump unit consists of a stacked series of rotating centrifugal impellers running on a central drive shaft inside a stack of stationary diffusers. The pressure increase is proportional to the number of stages while the pump volume increases as the diameter of the impeller increases.

Rotation of the impeller accelerates the liquid to be pumped which is then discharged into the diffuser where this kinetic energy is transformed into potential energy. The impeller/ diffuser pairs are arranged in series with the discharge of one unit being the suction of the next one. The number of pump stages may range between 10 to more than 100; depending on the pressure increase requirements.

The Protector or Seal unit

The Protector or Seal unit connects the drive shaft of the electric motor to the pump or gas separator shaft. It can act as an isolation barrier between well fluids, electric motor lubrication fluids and the electrical wiring. It is an expansion buffer for the motor oil which heats up when the ESP is initially run into the well and when the motor reaches operating temperature when the pump is operating. Also it helps the equalization of internal motor pressure and well annular pressure, and absorbs possible thrust generated by the pump.

Electric Motor

The electric motor is the "high tech" ESP component. It is powered by three phase alternating current, supplied from the surface by the cable connected to the motor at the pothead. Motors are available in sizes between 15 to 900 HP and two or even three motors may be placed in series to meet high pump power requirements.

The motor is filled with oil which insulates the electrical winding and the completion design must ensure that fluid will flow around the motor to provide sufficient cooling.
When the motor is switched off the head of fluid present in the tubing will reverse the flow direction through the pump as it flows back into the reservoir. This will cause the motor to spin backwards, so to restart the motor while it is rotating backwards must be avoided as the motor will burn out very quickly.

Sensor Package

A down hole sensor package may be mounted underneath the motor. The sensor sends valuable information to the surface as pump suction and discharge pressures, fluid intake temperature, electric motor temperature, motor and pumps vibration, electrical current leakage.

Gas Separator

A rotary gas separator may be included as part of the pump intake if there are high volume gas fractions as 20% to 40%. This is a centrifugal device which separates the lower density gaseous phase from the denser liquid phase. The latter is concentrated at the center of the device and enters the pump suction while the lighter; gas phase is expelled into the annulus. The gas rises to the wellhead where it is vented or collected in a low pressure, gas gathering system.

Pumping Head

Unlike positive displacement pumps, centrifugal pumps do not displace a fixed amount of fluid but create a relatively constant amount of pressure increase to the flow system. The output flow rate depends on backpressure. The pressure increase is usually expressed as pumping head, the equivalent height of freshwater that the pressure differential can support. In U.S. field units, the pumping head is expressed as:

$$h = \frac{\Delta p}{0.433} \tag{3.55}$$

where
$h =$ pumping head, ft
$\Delta p =$ pump pressure differential, psi

As the volumetric throughput increases, the pumping head of a centrifugal pump decreases and power slightly increases. However, there exists an optimal range of flow rate where the pump efficiency is optimum.

3.2.1.3 Basic Pump Selection

The pressure increase required from the pump to deliver is called “Total Dynamic Head (TDH)” and is the difference between the pump discharge and suction pressure. It is the sum of three components as shown in Figure 3.2.1.2.a.

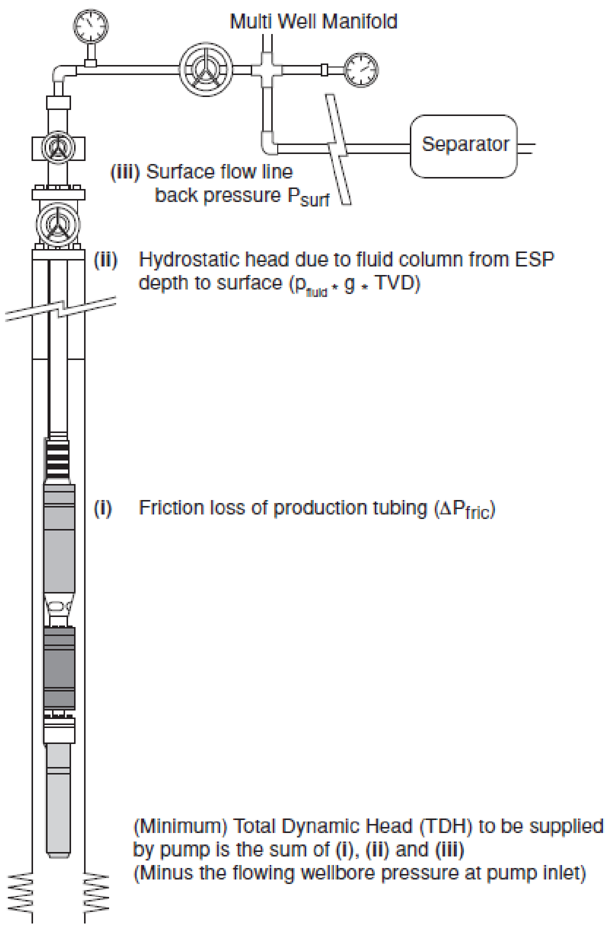


Figure 3.2.1.2.a: Pump duty requirements

- The hydrostatic head from the ESP pump to the surface. This is equal to the average density of the produced fluid in the tubing ρ_{fluid} multiplied by the True Vertical Depth (TVD) at which the ESP is installed and the acceleration due to gravity.
- The Friction pressure loss in the tubingΔP_{fric}.
- The surface pressureP_{surf} required to overcome flowline back pressure and flow the produced fluid to the separator at the required production rate. This can have a high value if the completion is a satellite well situated far away from the host platform.

TDH = (ρ_{fluid}*g*TVD) + ΔP_{fric} + P_{surf}

The data describing the performance of ESP's provided by the manufacture is normally measured with water. They also supply a correction factor, based on the actual density and viscosity of the fluids being pumped. Further correction is required if significant volumes of free gas are being injected by the pump.

One popular application area of ESP's is the production of viscous crude oils at high water cuts without formation of viscous emulsions. The design process can be simplified here since the density of the crude oil is similar to that of water, there is little gas and the produced fluid stream has an external water phase, so in that case the manufacturer's performance curves based on pumping water can be applied directly.

Once the pump has been chosen, optimum motor and seal section can be identified, along with the electric cable and variable speed drive. It also needs to be checked that the chosen combination will operate efficiently for a variety of well conditions (higher/lower well PI, greater water cut and lower reservoir pressure).

Mean Time before Failure/ Learning Curve

The choice of correct ESP design, along with the actual, operational installation of ESP’s, is a complex task. These results in the average run lifetime or “mean time before failure (MTBF)” often being very low initially when EPS's are first introduced into a producing area. The MTBF then increases as the “learning curve” is climbed. This is illustrated in Figure 3.2.1.4.b prepared from data presented at IQPC’s conference on Artificial Lift Equipment, Dubai 1997 and IQPC’s Artificial Lift Workshop, Aberdeen 1997. This figure shows how the THUMS project at Long Beach, California, has been employing ESP’s since 1965 and how the average run lifetime gradually increased throughout the 16 year history. During this period the numbers of operational ESP’s remained constant at about 600. ESP’s were only introduced into the North Sea in the late 1980’s. The initial run times were low compared to those achieved at THUMS, but a steep learning curve developed and by 1996 the average run lives had become similar.

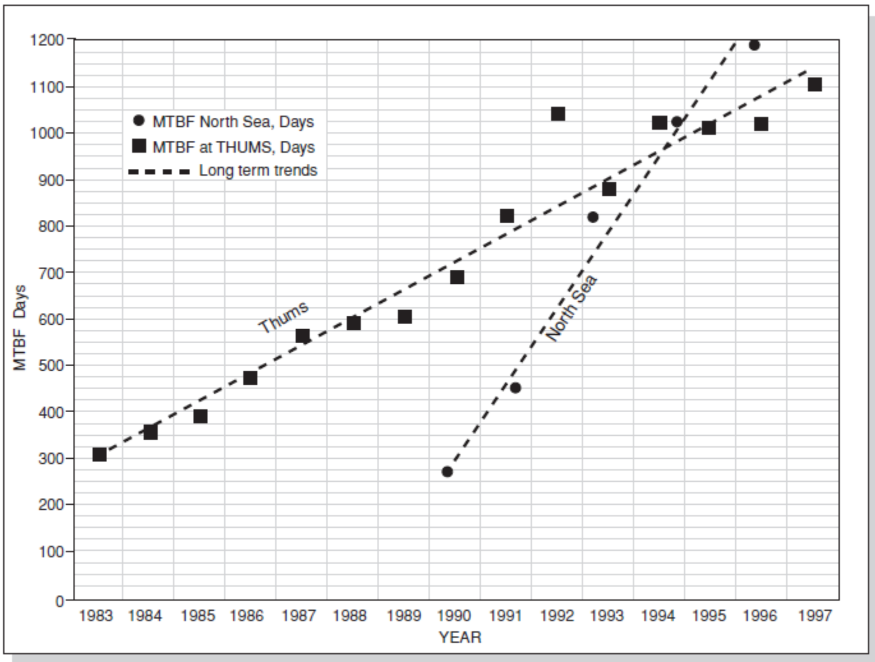


Figure 3.2.1.2.b: Average electric submersible pump lifetimes

The following procedure can be used for selecting an ESP:

1. Starting from well inflow performance relationship (IPR), the desirable liquid production rate q_{Ld} is determined. Then a pump size is selected from the manufacturer’s specification that has a minimum delivering flow rate q_{Lp}, that is, $q_{Lp} > q_{Ld}$.
2. From the IPR the flowing bottom-hole pressure p_{wf} is determined at the pump delivering flow rate q_{Lp}, not the q_{Ld} .

3. Assuming zero casing pressure and neglecting gas weight in the annulus, the minimum pump depth is being calculated by:

$$D_{pump} = D - \frac{p_{wf} - p_{suction}}{0.433\gamma_L}$$

where
D_{pump} = minimum pump depth, ft
D = depth of production interval, ft
p_{wf} = flowing bottom-hole pressure, psia
$p_{suction}$ = required suction pressure of pump,150–300 psi
γ_L = specific gravity of production fluid, 1.0 for freshwater

4. The required pump discharge pressure is determined based on wellhead pressure, tubing size, flow rate q_{Lp}, and fluid properties
5. The required pump pressure differential $\Delta p = p_{discharge} - p_{suction}$ and then required pumping head are calculated by Eq. (3.55)
6. From the manufacturer's pump characteristics curve, pump head or head per stage can be found. The required number of stages can then be calculated.
7. The total power required for the pump can be easily calculated by multiplying the power per stage by the number of stages.

ESP Classification

Like most down hole tools in the oil field, ESPs are classified by their outside diameter which vary from 3.5 to 10.0 in. The number of stages to be used in a particular outside diameter sized pump is determined by the volumetric flow rate and the lift requirements. Thus, the length of a pump module can be from 40 to 344 in. Electric motors are three-phase AC and can vary from 10 to 750 hp at 60 Hz or 50 Hz.

3.2.1.4 *ESP Applications*

The following factors are important in designing ESP applications:

- PI of the well
- Casing and tubing sizes
- Static liquid level

ESPs are usually for high PI wells and now days more ESP applications are found in offshore wells. The outside diameter of the ESP down-hole equipment is determined by the inside diameter (ID) of the borehole. There must be clearance around the outside of the pump down-hole equipment to allow the free flow of oil/water to the pump intake.

Typical ESP Applications

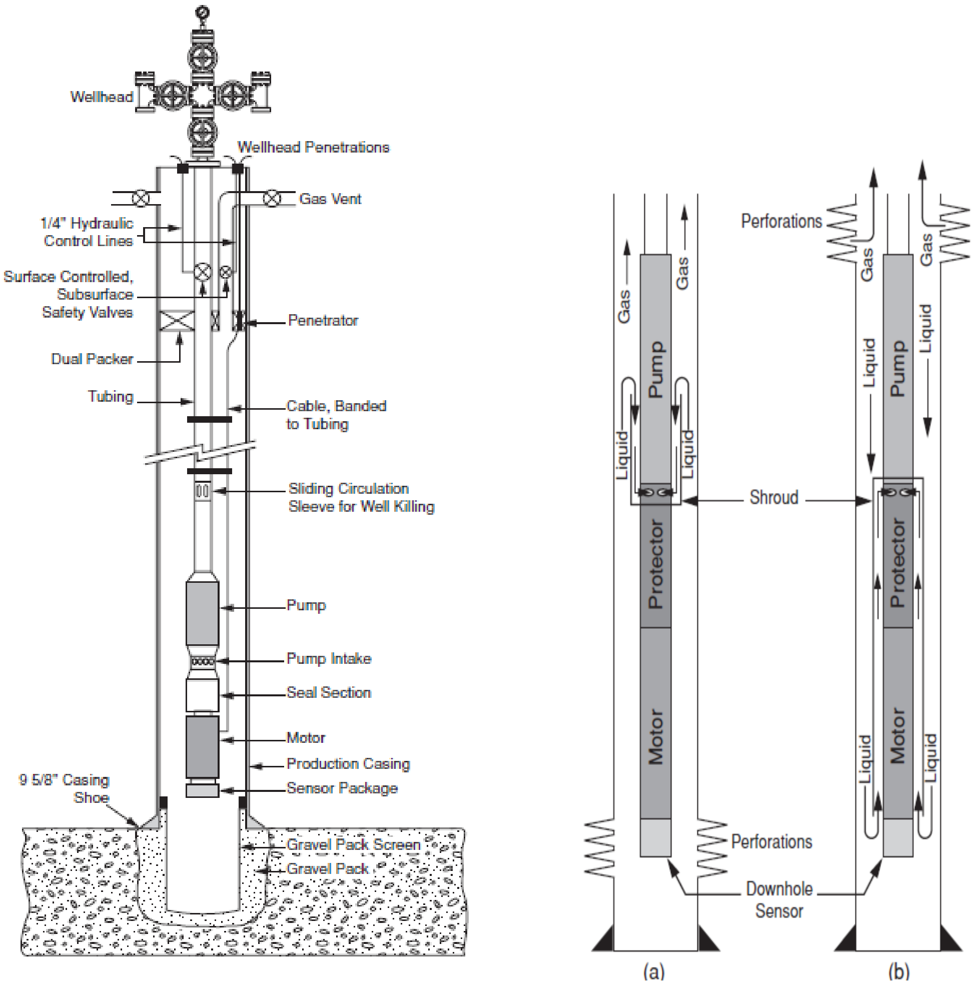


Figure 3.2.1.3.a: ESP completion incorporating packer and two surface controlled sub surface safety valves on the left and pump completion with shroud, pump completion under the perforations level on the right

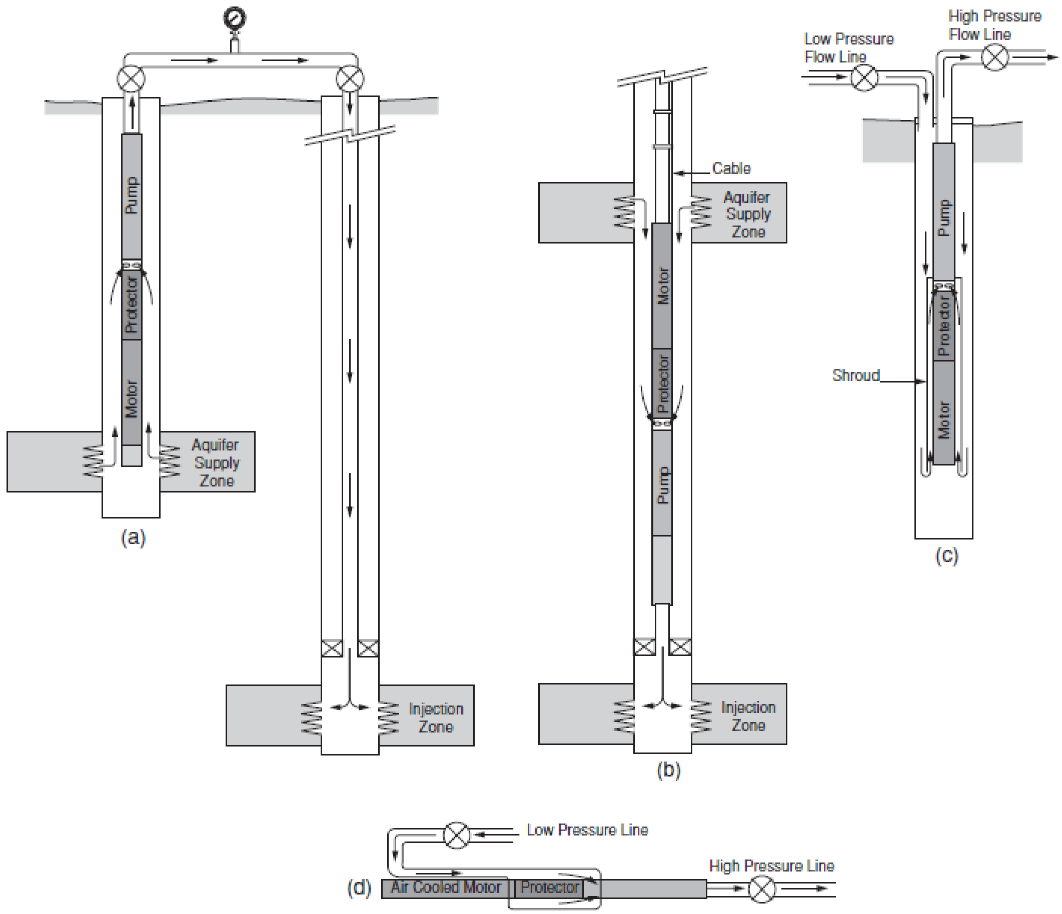


Figure 3.2.1.3.b: ESP applications (a) Direct water injection (b) Powered dumpflood with ESP (c) Pressure boosting surface pipelines with a shallow, subsurface mounted ESP (d) Horizontally mounted ESP surface pump

Horizontal Wells

ESPs are capable of producing large volumes of fluid and have design and operation flexibility. That makes them ideal candidates for production associated with horizontal wells.
Experience has shown that a high quality pump can be placed anywhere within the well at angles up to 80° providing the dogleg severity is not too great (< 6°/100 ft). Otherwise severe dog legs will result in bending of the ESP as it is pushed through this section of casing and this minor damage will lead to reduction in the pumps run-life.
Placing the pump near the bottom of the well maximizes the potential drawdown while the near horizontal section will enhance the separation of the gas to the upper portion of the wellbore due to its lower density.

"Y" Tool

The "Y" tool is a device that allows wireline or coiled tubing accessing below the ESP (Figure 3.2.1.3.c). The bypass tubing should be at least $2\ {}^{3}/_{8}$ in OD allowing $1\ {}^{11}/_{16}$ in, logging tools to pass. However, the larger diameter tubing reduces the maximum diameter of ESP that can be installed.
Installation of "Y" tools allows all normal wireline and coiled tubing operations to be carried out below the ESP:

- Cased hole logging
- Well stimulation
- Perforating
- Setting bridge plugs for water shut off
- Installation and recovery of pressure memory gauges
- Running and retrieval of plugs
- Downhole sampling

Omission of the “Y” tool from the downhole completion design implies that these operations are only possible when tubing and ESP are recovered to the surface.

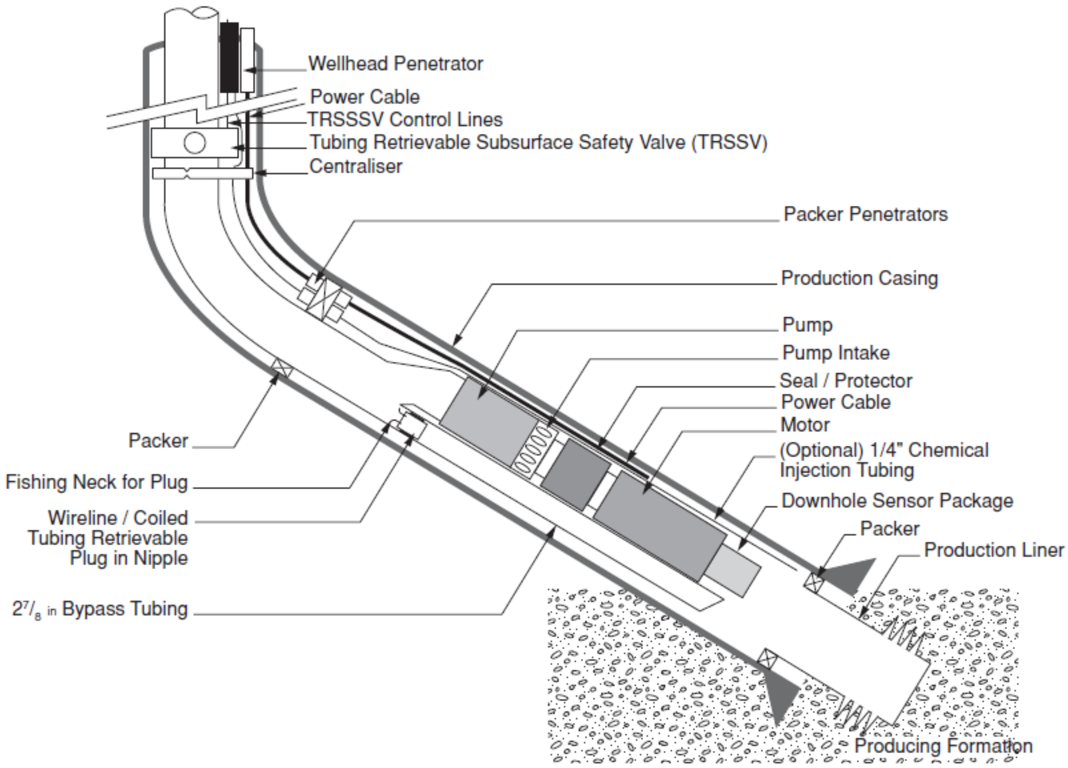


Figure3.2.1.3.c: The "Y"tool"

3.2.1.5 Electric Submersible Pump Performance

The centrifugal pump unit employed in ESP’s is a dynamic displacement pump in which the pump rate depends in the pressure head generated. The pump rate is low when the pressure head is high and vice versa.This is different from the positive displacement pumps discussed earlier in which the pump rate and discharge pressure are independent of one another.
The relationship between pump rate and pressure generated for dynamic displacement pumps is the pump characteristic curve and it is measured by the pump manufacturer in laboratory tests using a standard fluid (water) with the pump running at 3500 rpm (60 Hz electrical supply) or 2915 rpm (50 Hz supply).

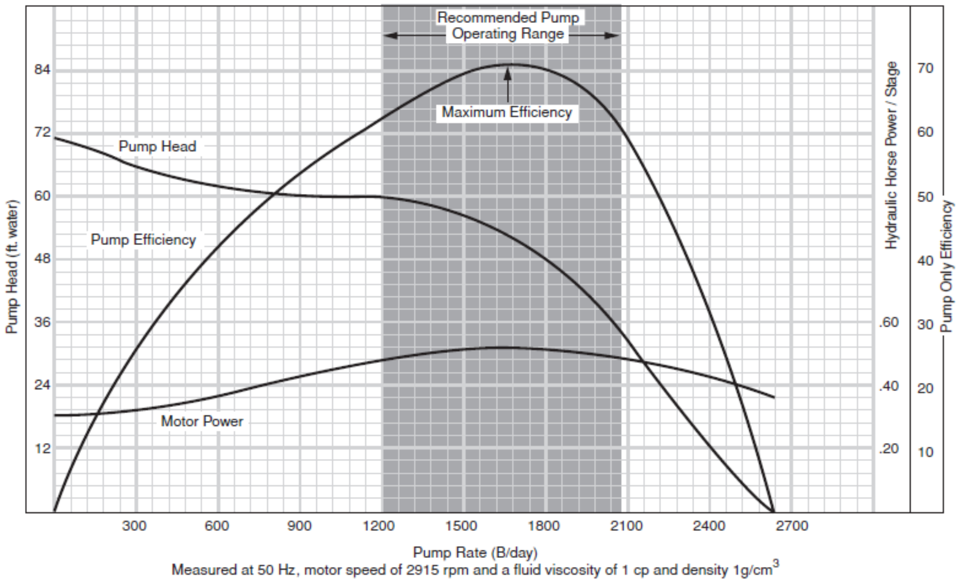


Figure 3.2.1.7: A typical pump characteristic curve for a centrifugal pump

The "pump head" or increase in pressure per stage ΔP, is expressed in terms of the pressure generated by an equivalent column of water H_{water}. It decreases as the pump rate increases:

$$\Delta p = \rho_w * g * TVD = 0.433 * \gamma * H_{water}$$

The discharge pressure is proportional to the specific gravity γ for liquids with the same viscosity. The pump head/ pump rate relationship can be used for all liquids, though it requires correction for changes in viscosity.

Pump power is the work done per unit time and equals the pump rate multiplied by the pump head q*ΔP. Power is expressed in terms of Kilowatt (KW) or Horse Power (HP); 1 HP = 0.746 KW. The pump power or hydraulic power is the useful work done by the pump; while the mechanical power is the work done by the electrical motor which is driving the pump. The pump efficiency E is:

$$E = \frac{\text{hydraulic power}}{\text{mechanical power}}$$

Pump Efficiency is also recorded in the pump characteristic curve and it is recommended that the pump operating range must be within ±10% of the maximum efficiency point.

3.2.1.6 Monitoring the Performance of Electric Submersible Pumps

Information on well performance can be derived by a 24 hour chart which records the electrical current consumed by an electric submersible pump during normal operation (Figure 3.2.1.5.a) or when the well is being pumped off and shows a much more erratic behavior (Figure 3.2.1.5.b).

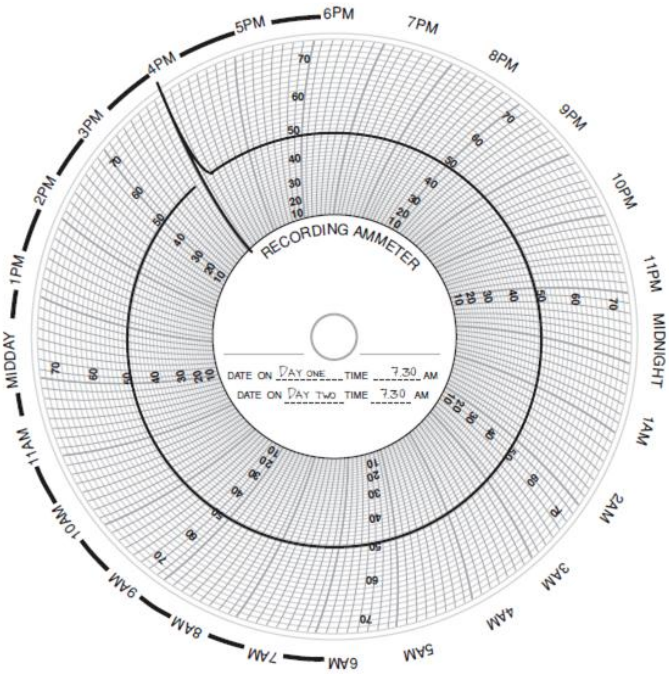


Figure 3.2.1.5.a: Ammeter chart monitors electric submersible pump performance at normal operation

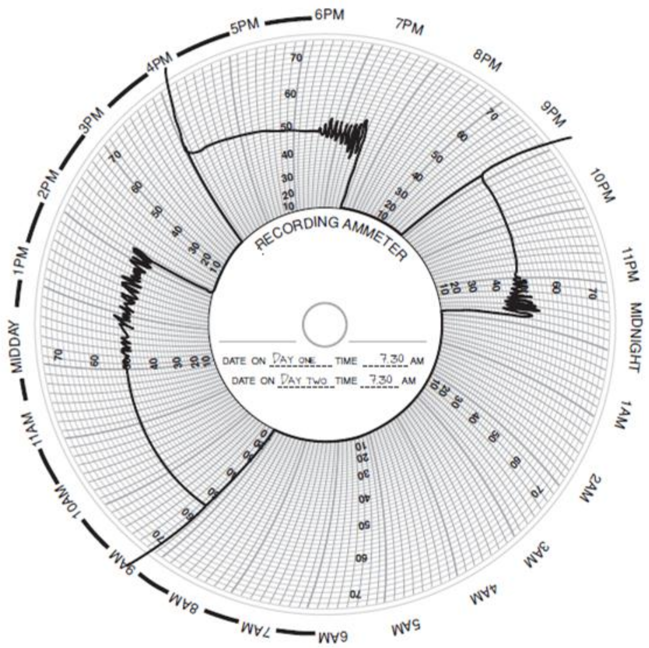


Figure 3.2.1.5.b: Ammeter chart monitors electric submersible pump performance when well is pumped off

Description of chart monitoring ESP performance when well is pumped off

The pump started at 09.15 am as shown by the large, initial surge in current while the motor is getting “up to speed”. This occurs because the initial resistance of the motor winding is low. As the motor speed increases the inductive resistance increases rapidly. A steady current is then drawn for the next 3 hours and decreasing slightly as the fluid head above the pump also decreases.

At 12.10 the current begins to oscillate rapidly and the size of these oscillations increases until 2.15 pm when the pump was shut down. It was suspected that the problem was due to gas being formed when the flowing bottom hole pressure was reduced below the bubble point, leading to gas locking. This was confirmed by leaving the fluid level in the well to build up for 100 minutes and restarting the pump at 4.05 pm.

The same cycle repeats itself, however this time the problems appear after some 2.5 hours of steady production. The pump was shut down for the second time at 7.15 pm.
A third cycle was started at 8.50 pm after a second 100 minute shut in and current oscillation starts again after 2 hours of production. The well was shut in just after 11.30 pm.

The basic problem is that the ESP is pumping fluid to the surface faster than the fluid is flowing into the well from the reservoir. Continual stopping and restarting the ESP motor is not recommended due to damage to the motor winding by the initial high current surge as the motor begins to rotate. This will lead to early motor burnout. There are three options:

- Install a lower capacity (smaller) pump section
- Operate the pump at a lower speed using a VFD
- Stimulate the well to improve the inflow performance

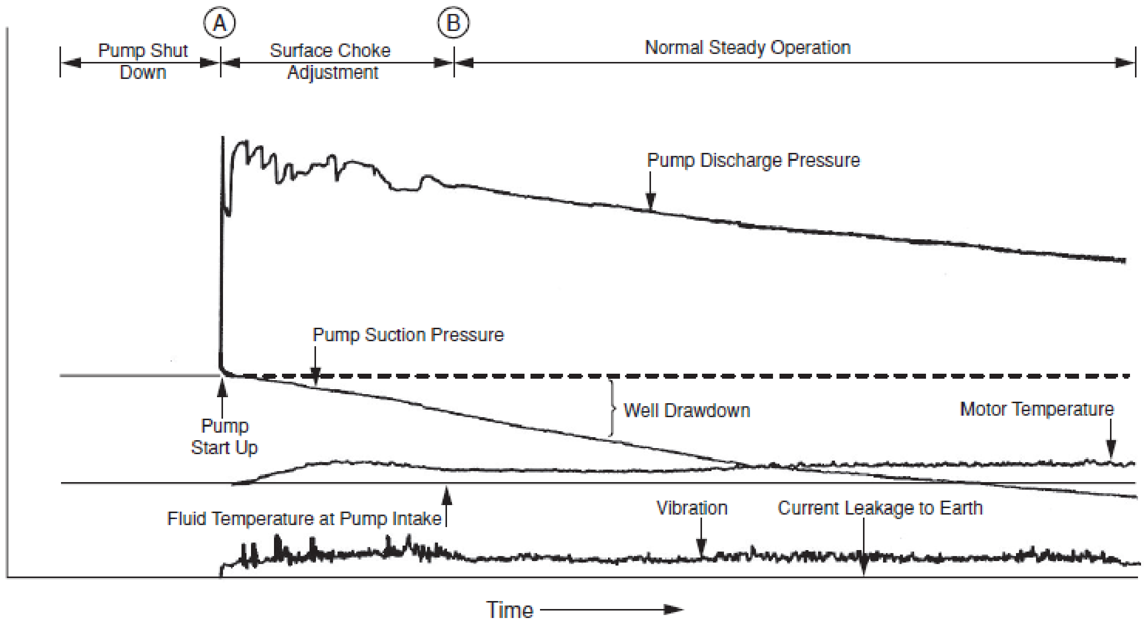


Figure 3.2.1.5.c: Electric submersible pump and motor condition monitoring

Before energizing the pump, the intake and discharge pressures have the same value. The pump starts up at point A, as shown by (Figure 3.2.1.5.c):

- The pump discharge pressure increases
- The motor temperature becoming warmer than the fluid entering the pump
- A limited amount of vibration during surface choke adjustment

A period of surface choke adjustment follows, as shown by fluctuations in the pump discharge pressure and increased vibration. After point B, steady operating conditions are achieved and a slow decline in pump suction and discharge pressure are observed as the well's flowing bottom hole pressure is reduces.

Protection of the ESP can now be achieved by monitoring the pump's condition and shutting it down when problems develop before physical damage to the pump results. Thus "pump off" control can be implemented by stopping the pump when the intake pressure drops below a preset value and the pump is then restarted once the well pressure builds up to a second, higher predetermined value.

3.2.1.7 Advantages and Disadvantages of Electric Submersible Pumps

Advantages

- Can be installed in deviated wells ($< 80°$)
- High production rates
- Controllable production rate
- Efficient Energy usage (>50% possible)
- Access below ESP via "Y" tool
- Comprehensive down hole measurements available
- Can pump against high Flowing-Tubing Head Pressure
- Low surface profile for Urban and offshore environments
- Quick restart after shut down
- Concurrent drilling and production safer compared to gas lift
- Long run pump life possible

Disadvantages

- Susceptible to damage during completion installation
- Tubing has to be pulled to replace pump
- Not suitable for low volume wells (<150 bpd)
- Pump susceptible to damage by produced solids (sand/scale/asphaltene)
- High GOR's presents gas handling problems
- Power cable requires penetration of well head and packer integrity
- Viscous crude reduces pump efficiency
- High temperatures can degrade the electrical motors

3.2.2 Hydraulic Jet Pumping

The pump converts the energy from the injected power fluid (water or oil) to pressure that lifts production fluids. Dirty and gassy fluids present no problem to the pump, as there are not any moving parts involved. The jet pumps can be set at any depth as long as the suction pressure is sufficient to prevent pump cavitation problem. A hydraulic jet pump installation is shown in Figure 3.2.2.

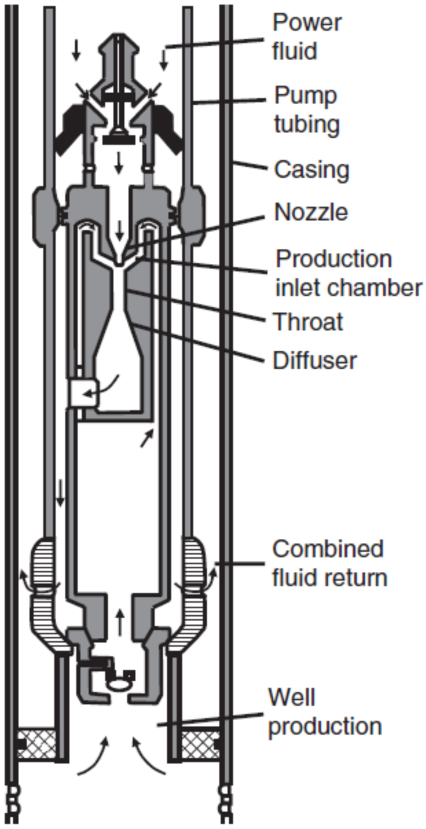


Figure3.2.2: Sketch of a hydraulic jet pump installation

3.2.2.1 Working Principle

Figure 3.2.2.2 illustrates the working principle of a hydraulic jet pump. It is a dynamic-displacement pump that differs from a hydraulic piston pump in the manner in which it increases the pressure of the pumped fluid with a jet nozzle. The power fluid enters the top of the pump from the injection tubing. The power fluid is then accelerated through the nozzle and mixed with the produced fluid in the throat of the pump. As the fluids mix, the momentum of the power fluid is partially transferred to the produced fluid and increases its kinetic energy (velocity head).

Some of the kinetic energy of the mixed stream is converted to static pressure head in a carefully shaped diffuser section of expanding area. The fluid mixture in the annulus can be lifted to the surface if the static pressure head is greater than the static column head in the annulus.

3.2.2.2 Technical Parameters

The nomenclatures in Fig. 3.2.2.2 are defined as:

p_1 =power fluid pressure, psia
q_1 =power fluid rate, bbl/day
p_2 =discharge pressure, psia
$q_2 = q_1 + q_3$, total fluid rate in return column, bbl/day
p_3 = intake pressure, psia

q_3 = intake (produced) fluid rate, bbl/day
A_j =jet nozzle area, in^2
A_s =net throat area,in^2
A_t = total throat area, in^2

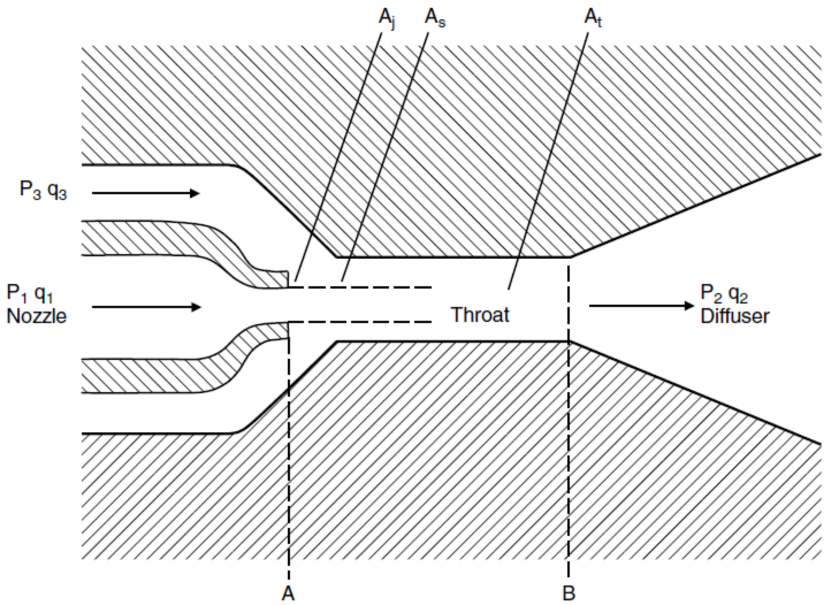


Figure 3.2.2.2: Working principle of a hydraulic jet pump

The following dimensionless variables are also used in jet pump literature (Cholet, 2000):

$$R = \frac{A_j}{A_t}$$

$$M = \frac{q_3}{q_1}$$

$$H = \frac{p_2 - p_3}{p_1 - p_2}$$

$$\eta = MH$$

where
R = dimensionless nozzle area
M = dimensionless flow rate
H =dimensionless head
η =pump efficiency

3.2.2.3 Selection of Jet Pumps

Selection of jet pumps is made on the basis of manufacturer's literatures where pump performance charts are usually available. Figure 3.2.2.3 presents an example chart that shows the effect of M on H and η. For a given jet pump specified by R value, exists a peak efficiency η_p. It is good field practice to attempt to operate the pump at its peak efficiency. If M_p and H_p are used to denote M and H at the peak efficiency, respectively, pump parameters should be designed using

$$M_p = \frac{q_3}{q_1} \qquad and \qquad H_p = \frac{p_2 - p_3}{p_1 - p_2} \tag{3.56}$$

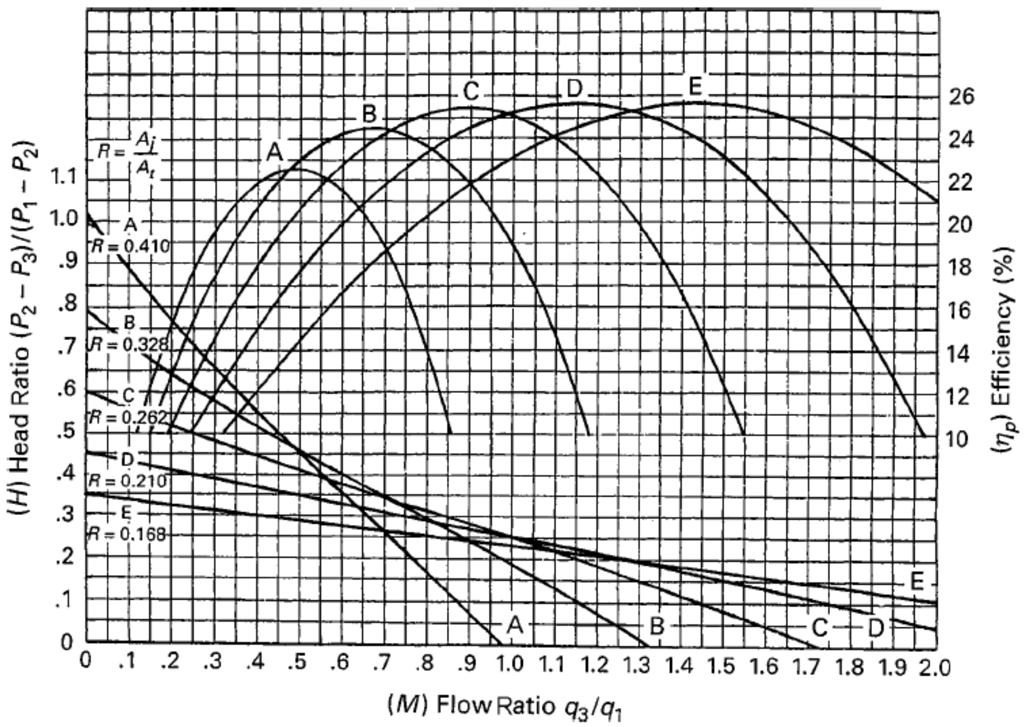


Figure 3.2.2.3: Example jet pump performance chart

Where M_p and H_p values can be determined from the given performance chart. If the H scale is not provided in the chart, H_p can be determined by

$$H_p = \frac{\eta_p}{M_p}$$

The power fluid flow rate and pump pressure differential are related through jet nozzle size by

$$q_1 = 1214.5 A_j \sqrt{\frac{p_1 - p_3}{\gamma_1}} \tag{3.57}$$

where γ_1 is the specific gravity of the power fluid, q_1 in bbl/day, and p_1 and p_3 are both in psi.

The following procedure can be taken to select a jet pump:

1. The desired production rate of reservoir fluid q_3 based on well IPR is selected and the required bottom hole pressure p_{wf} is determined.
2. The pump setting depth D is designed and an estimation about the pump intake pressure p_3 based on p_{wf} and flow gradient below the pump is required.
3. From manufacturer's literature, a pump is selected with R value.The M_p and H_p values for the pump based on pump performance curves should be determined.
4. Power fluid rate q_1 is calculated by:

$$q_1 = \frac{q_3}{M_p}$$

5. Based on tubing flow performance, the calculations about the required discharge pressure $p_{2,r}$ are using the production rate $q_2 = q_1 + q_3$.
6. The power fluid pressure p_1 required to provide power fluid rate q_1 with Eq. (3.57), is calculated by:

$$p_1 = p_3 + \gamma_1 (\frac{q_1}{1214.5 A_j})^2$$

7. The available discharge pressure p_2 from the pump with Eq. (3.56), that is:

$$p_2 = \frac{p_3 + H_p p_1}{1 + H_p}$$

8. If the p_2 value is greater than $p_{2,r}$ value with a reasonable safety factor, the chosen pump is okay to use, so the nest is Step 9. Otherwise, Step 3 must change and a different pump must be selected. If no pump meets the requirements for the desired production rate q_3 and/or lifting pressure $p_{2,r}$, Step 2 must consider and a change to pump setting depth must be done or a reduction to the value of the desired fluid production rate q_3.
9. The required surface operating pressure p_s based on the values of p_1 and q_1 and single-phase flow in tubing must be calculated.
10. Input power requirement are calculated by:

$$HP = 1.7x10^{-5} q_1 p_s$$

where
$HP =$ required input power, hp
$p_s =$required surface operating pressure, psia

3.2.2.4 Advantages and Disadvantages of Hydraulic Jet Pumps

Advantages

- Can lift from as deep as 20,000 feet
- Can produce 25,000 B/D from 5,000 feet
- Crooked holes present no problems
- Unobtrusive in urban locations and use on offshore platforms
- Power source can be remotely located
- Down hole pumps can be installed /retrieved using the power fluid
- Can use well’s produced fluids for power fluid (water or oil)
- Power fluid can be heated to reduce viscosity of produced fluid
- Inhibitors can be mixed with the power fluid for controlling corrosion, scale, emulsions

Disadvantages

- Total system efficiency approximately 10-30%.
- Pump will cavitate if more production than planned is forced through the pump
- Power oil systems are a possible fire issue
- High surface power fluid lines are required
- Any leaks when using power oil pose an environmental issue
- Qualified personnel needed for trouble shooting in field

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